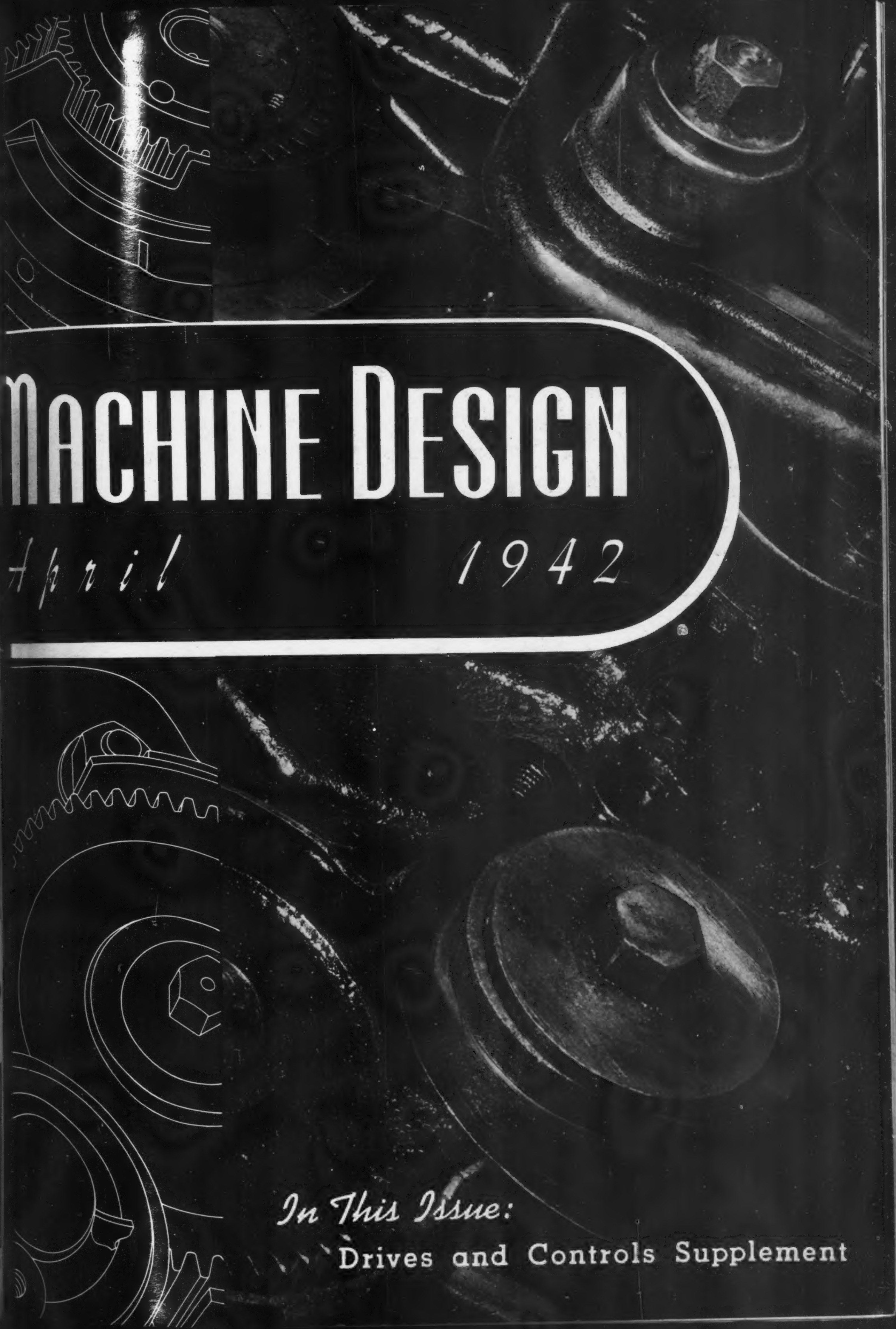

MACHINE DESIGN

April 1942

In This Issue:

Drives and Controls Supplement


MISTER,
we keep 'em Clean!
Keeping things shipshape is an unending job in the navy . . . for dust settles on horizontal surfaces everywhere whether these surfaces are parts of a boat far at sea or parts inside motor control in any factory.
Because dust *is* found in even the most unexpected places, it is not surprising to find it *inside* Motor Control equipment in *any* busy factory. Cutler-Hammer engineers recognized this . . . and they also knew that dusty, dirty contacts meant trouble . . . so they designed their famous VERTICAL contact Motor Control. *Vertical* contacts simply can't collect dust and dirt. They *stay* clean to work better and last longer. Such Motor Control that needs less attention and sets records for dependable performance in these days and nights of continuous high-speed production contributes not only to industrial efficiency but directly to national security. CUTLER-HAMMER, Inc., 1310 St. Paul Ave., Milwaukee, Wisconsin. *Associate:* Canadian Cutler-Hammer, Ltd., Toronto.
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Cutler-Hammer Vertical Contacts are the mark of better Motor Control, another extra dividend on Cutler-Hammer's unequalled specialized experience and decades of Motor Control leadership.
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MACHINE DESIGN

THE PROFESSIONAL JOURNAL OF CHIEF ENGINEERS AND DESIGNERS

Volume 14 APRIL, 1942 Number 4

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MAIN OFFICE: The Penton Publishing Co., Penton Bldg., Cleveland, O.

BRANCH OFFICES: *New York,* 110 East 42nd St.; *Chicago,* 520 N. Michigan Ave.; *Pittsburgh,* Koppers Bldg.; *Detroit,* 6560 Cass Ave.; *Washington,* National Press Bldg.; *San Francisco,* 1100 Norwood Ave.; *London,* 2 Caxton St., Westminster, S.W.1

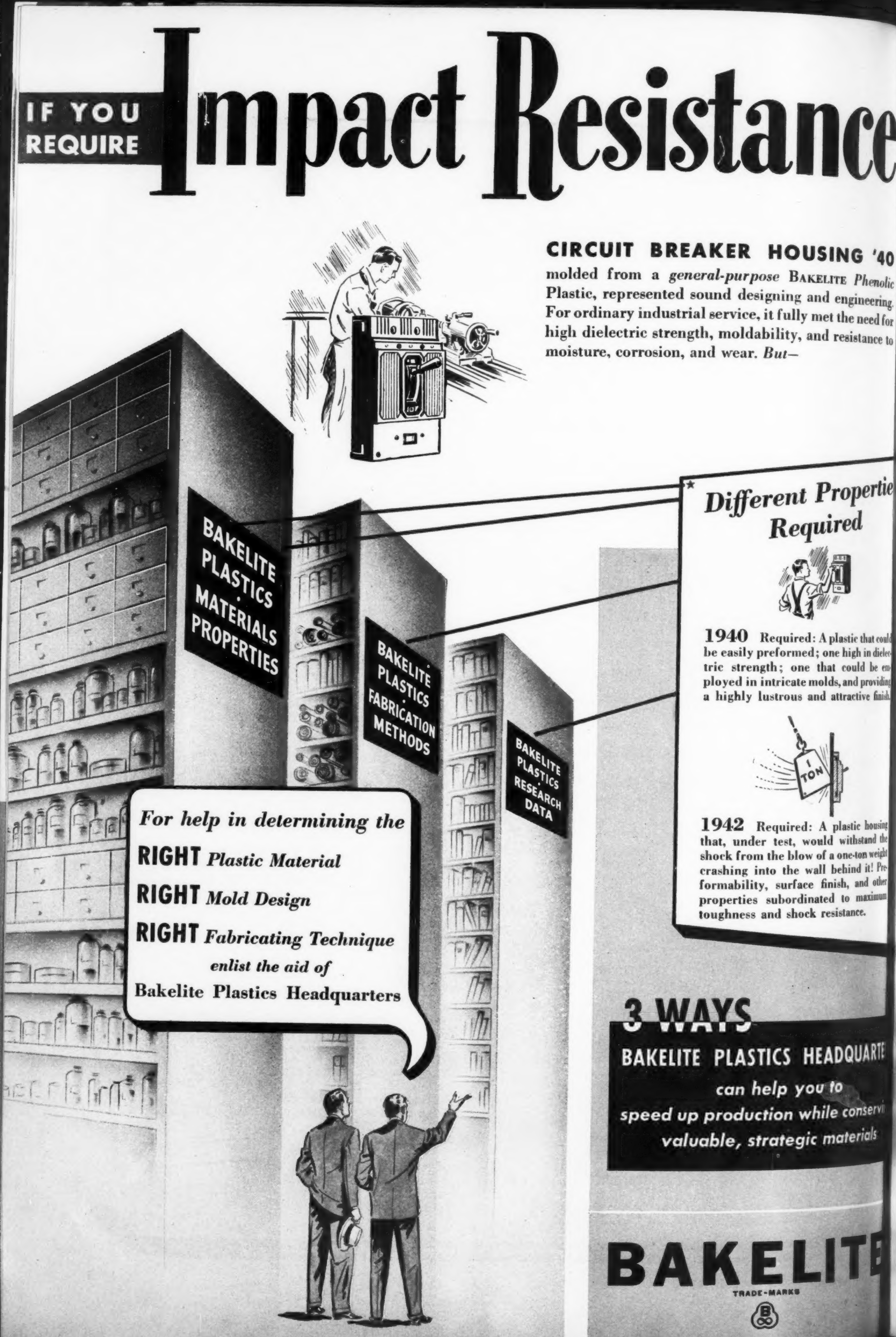
PUBLISHED BY The Penton Publishing Co. E. L. Shaner, Pres. and Treas.; G. O. Hays, Vice Pres.; F. G. Steinebach, Secy. Published on seventh of month. Subscription in U.S. and possessions, Canada, Cuba, Mexico, Central and South America: Two years, $10; one year, $6. Single copies, 50 cents. Other countries: Two years, $14; one year, $8.

Acceptance under act of June 5, 1934, authorized July 20, 1934.

Itemized Index

Classified for Convenience when Studying Specific Design Problems

MACHINE DESIGN is indexed in Industrial Arts Index and Engineering Index Service, both available in libraries generally.



IF YOU REQUIRE Impact Resistance
CIRCUIT BREAKER HOUSING '40
molded from a *general-purpose* BAKELITE *Phenolic* Plastic, represented sound designing and engineering. For ordinary industrial service, it fully met the need for high dielectric strength, moldability, and resistance to moisture, corrosion, and wear. *But—*
BAKELITE PLASTICS MATERIALS PROPERTIES
BAKELITE PLASTICS FABRICATION METHODS
BAKELITE PLASTICS RESEARCH DATA
*Different Propertie Required
1940 Required: A plastic that could be easily preformed; one high in dielec tric strength; one that could be em ployed in intricate molds, and providing a highly lustrous and attractive finish.
1 TON
1942 Required: A plastic housing that, under test, would withstand the shock from the blow of a one-ton weight crashing into the wall behind it! Pre formability, surface finish, and other properties subordinated to maximum toughness and shock resistance.
For help in determining the
RIGHT *Plastic Material*
RIGHT *Mold Design*
RIGHT *Fabricating Technique*
enlist the aid of
Bakelite Plastics Headquarters
3 WAYS
BAKELITE PLASTICS HEADQUARTE
can help you to speed up production while conservi valuable, strategic materials
BAKELITE
TRADE-MARKS

Topics

INDUSTRY must find a way to do the job of conversion to war production; it must not make the mistake of relying on government, according to Donald Nelson. "Industry's responsibility is great. The job will take brains and initiative . . . but we can do it if we stop thinking about what we're going to do to the enemy in 1943 and start thinking about what we are going to do to him now."

♦

FOR every 700 automobiles *not* made, enough rubber and steel are saved to produce 30 twenty-seven ton medium-size tanks; enough tin to coat 700,000 cans in which to put food for soldiers and sailors; enough aluminum to make one fighter plane; enough nickel to produce 70,000 pounds of nickel steel for armor plate and armor-piercing bullets; enough zinc and copper to make 1,680,000 brass cartridge cases for .30 caliber ammunition. These materials are necessary to make the weapons we must have before we can have victory.

♦

CONSERVATION of welding electrodes through joint design, close fitting, and maximum utilization of welding rods is recommended by the Lincoln Electric Co. to aid in the war effort. Electrodes best suited for each job should be selected. They should be fast flowing, have proper physical characteristics, produce a flat bead, be of sufficiently large diameter, and give a minimum of spatter.

♦

SOME small parts which have been cast of gray iron involving melting, casting and machining are now made from scrap steel chips that are crushed in a ball mill, cold formed, sintered, and hot pressed to shape. Virgin material and skilled labor are thus released for war effort.

♦

ACCORDING to Charles F. Kettering, "Research has come to be surrounded by a good deal of meaningless mystery. Research, after all, is nothing but prospecting for knowledge. The world isn't finished. The day of discovery has not passed. There will always be new frontiers."

PRODUCTION of half an airplane at the push of a button is envisioned by Paul H. Merriman, chief engineer of Glenn L. Martin Co., making possible the production of 1000 planes a day. He says, "Sixty spot welds could be made in the time it now takes to install one rivet. With the perfection of aluminum spot welding, this will soon be possible."

♦

WHERE special parts are indicated in a machine it is well to consider redesign to utilize standard commercial parts, especially during these times. Such a program, according to New Departure Division, may have far-reaching results. "One special may well delay the delivery of many standard bearings, as well as the machines awaiting these bearings. Designers should select and specify standard bearings while the job is on the board. Result: More bearings. More machines. More armament. More hell for the Japs and their partners in crime."

♦

GOOD criterion of the ability of parts manufacturers to meet the emergency is the production index of the American Gear Manufacturers association. Industrial gear sales for February were 34.7 per cent above a year ago and 22.5 per cent above the preceding month.

♦

ON TEST, big Buick bomber engines produce noise levels of 135 decibels, a volume beyond the capacity of human endurance. By locating the engines behind thick concrete and insulated walls, test men have only 57 decibels hurled against their ears which is neither a harmful nor uncomfortable volume.

♦

SURVEYS of Willys-Overland Motors disclose that a large percentage of America's 40,000 automobile dealers can share in war production by taking subcontracts for their trained mechanics and machine equipment. Suitable for light manufacturing operations, lathes, shapers, grinders, drills, milling machines, air compressors and other equipment are available in repair shops in large numbers.

Conversion for War—Design's Great Opportunity!

By Kenneth D. Moslander

Engineering department designed for maximum efficiency is devoted exclusively to war work on airplane engines

PRIME requisite of the successful prosecution of the war is conversion. This refers to the changeover of all our resources, the exertion of all our efforts in the common cause. It not only means changing over plants to production for war, but far more than that. To the engineer it means a refocussing of his talents on problems arising out of the emergency. It means constructive thinking along the lines of new and more deadly war machines; it means the application of technical knowledge and ability directed toward making our present equipment outstandingly superior; it means hard work and lots of it!

On the shoulders of engineering management rests the job of facilitating, to the greatest possible extent, the execution of this work. Is all possible equipment available for the most efficient utilization of designers' time? Drafting machines, printing machines, copying machines, etc., are to the designer what the Garand is to the soldier. Production problems exist even in the design department; the more and better the machines available, the faster the work will be done.

Many designers possess demonstrated inventive ability, but this is not limited to the design of complete machines, civil or military. Shown in *Fig.* 2 is a boring machine that, in the days of automobiles, had four spindles for boring Ford V-8 engine blocks. Intensive engineering effort followed by three weeks in a tool and die shop resulted in its conversion to a three-spindle machine for boring holes in a 75-millimeter tank gun cradle. Parallel instances are innumerable in which existing machine tools, long used in peacetime industry and superficially considered to have no utility in

the war program, have been redesigned and rebuilt into versatile, precision, armament-producing equipment. Every chip cut by every machine which can be commandeered by the war effort is a chip from the hide of the axis. No one knows better the potentialities of the machine tool equipment in a given plant than the engineer who has been, for years, designing work to be done on those machines. Can these machines, by redesign, be mobilized 100 per cent for war?

Important as is the redesign of existing production equipment, development and redesign of military materiel is by far the primary need. Most fruitful of the opportunities are for different, faster or more efficient methods of manufacture

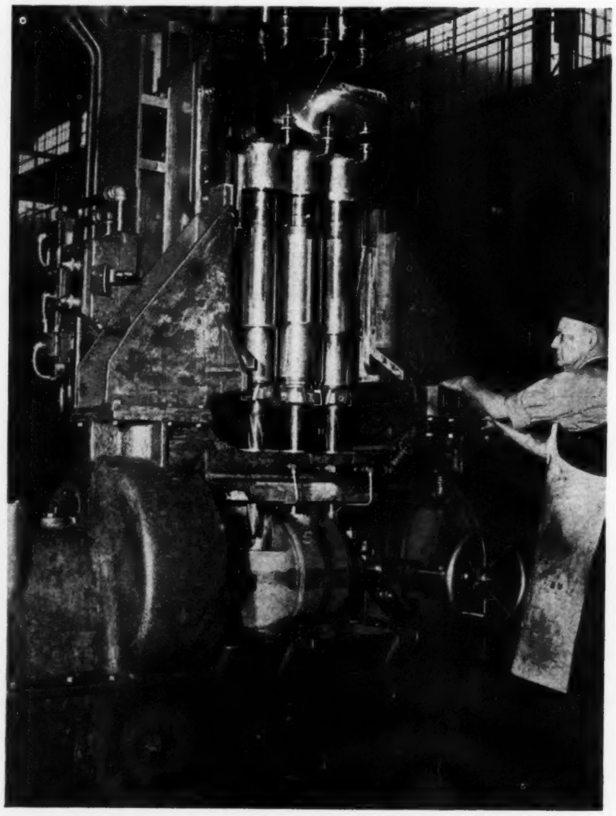


Fig. 2—Four-spindle boring machine has been redesigned to a three-spindle machine for boring holes in tank gun cradles

as well as the development of designs which make more sparing use of critical materials. Field gun and tank parts, formerly forged or cast have been replaced successfully with stamped parts, interchangeable with the original. Centrifugal casting is finding broad application in war equipment; rifling grooves in gun barrels are now being broached. New low-alloy steels combined with newly developed heat treating procedures are effecting important economies of alloying elements. The engineer knows these things. Is the utmost being done to utilize this knowledge to the maximum of effectiveness?

First major industry in this country to be converted exclusively to war work, and one in which engineering knowledge is being used to the maximum extent, is the automotive. The tremendous job which the former automobile plants are doing stands as a monument to engineering skill and should serve as a challenge to designers and technical men throughout the country.

In this industry, selfish barriers are down; the present engineering personnel exceeds in number the normal peacetime staff. Interchange of not only ideas but also of specialized engineers is encouraged both among different divisions of the same company and among the several companies in the industry.

These technical men do not rest at translating military designs to the production line. They are constantly seeking improvements which are submitted to military officials for approval as fast as they are developed. They are applying the "automotive know how" to war equipment, passing along accepted and proved fabricating techniques so that time can be saved and production expedited.

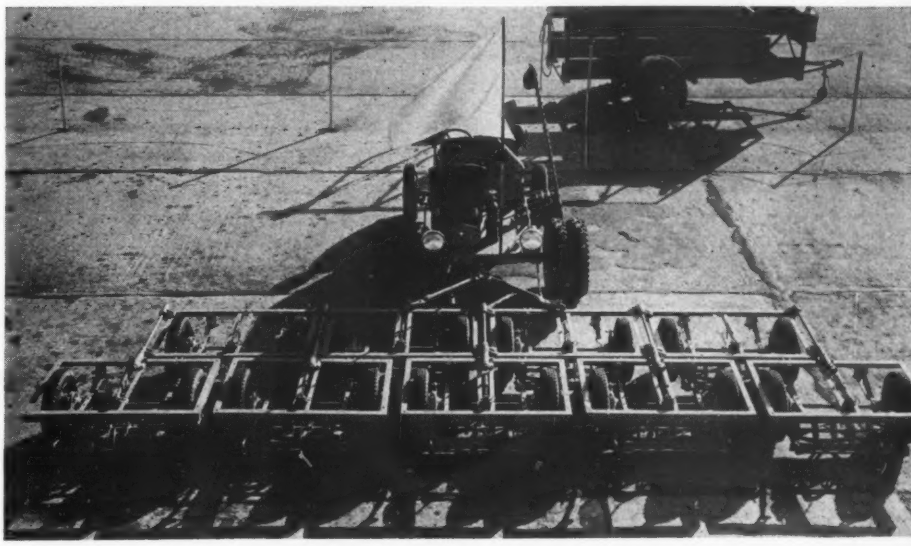
Other engineering groups from the motor companies are out in the field with the armed forces, studying their present equipment and analyzing their needs for new types. Data so obtained are taken back to the engineering department and new designs developed for submission to the armed forces. They are then put to the test and finally approved for production.

Method Illustrated by Example

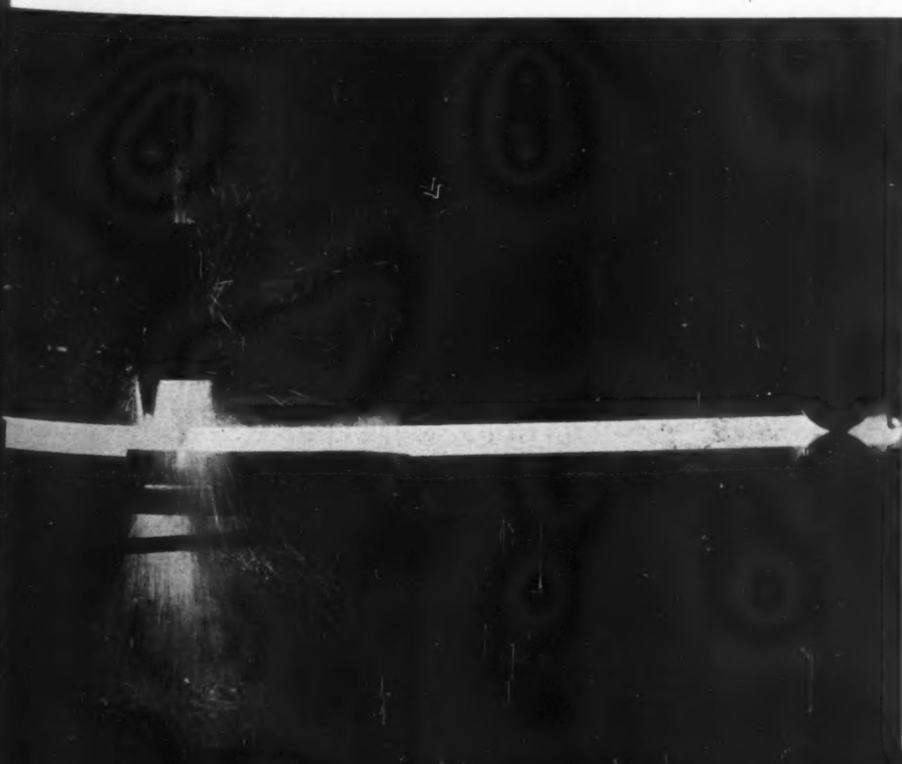
Take hypothetical case and see how it works. Automotive engineers watch military maneuvers and consult with the generals directing them. Someone offers a suggestion that an armed combat car, mounting an antitank gun, several machine guns and antiaircraft guns, would be able to supplement the work of a tank. Being a wheeled vehicle, it would have greater mobility, speed and perhaps even greater fire power. So engineers take this idea back to their drafting boards and, starting with the basic element of a motor truck, proceed to evolve a brand new type of combat car, even going so far as to draw up a dozen different ideas in such vehicles of which, after careful consideration, the best is selected.

These are some of the methods—there are others. Our engineers have the will and ability. As a nation we are faced with the absolute necessity to do this job of work as we have done others under far less stimulus. We can be confident that our engineers will "deliver the goods". In the words of Donald Nelson, "If we are to achieve victory . . . . then we on the production lines must abandon every other consideration except increasing production and increasing it every day. If we fail in that, we shall burn in the flames of a public wrath so intense that in its heat it might consume the very standards we have set for free men to live by".

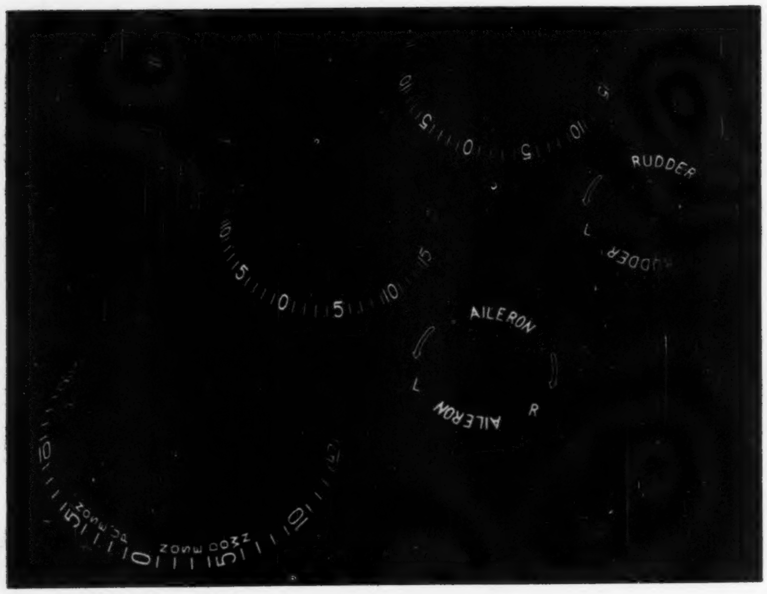
Scanning the Field for IDEAS



Grass blitzer developed by Worthington Mower Co., right, does its part in keeping turf on airfields in good condition, yet is economical in operation and requires no additional facilities for travelling from one field to another. Consisting of a gang of nine mowers pulled by a tractor, each mower unit is hinged transversely to follow irregularities in the terrain. Hinges do not act longitudinally because wheels of the front and rear rows automatically set the mower height between short distances. Gang is easily dismantled by one man by uncoupling the hinges, and may be loaded into the trailer unit shown in the background. In this way the mower furnishes its own transportation over highways at thirty miles an hour and relieves men and trucks for more pressing tasks.

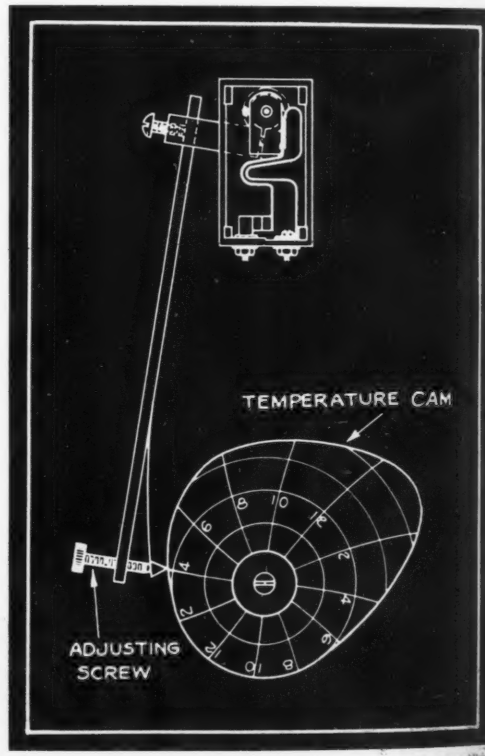


Cutting pipe to length, left, in a continuous tube mill requires accurate control of saw carriage travel and cutoff. Resembling cutter action in a cigarette machine, the saw oscillates back and forth along the pipe and at the instant of cutting is travelling at the speed of the pipe which is 400 feet a minute. Electrically synchronized control automatically operates the cutoff so that the cut is made with the required two inches of exact length.

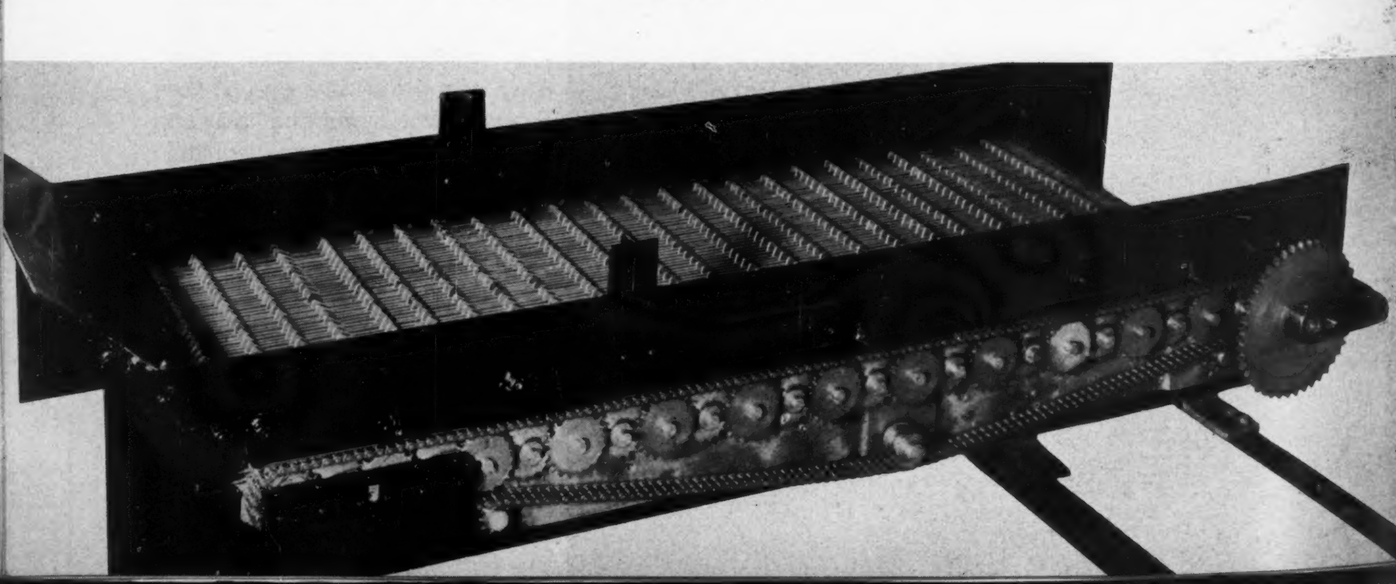


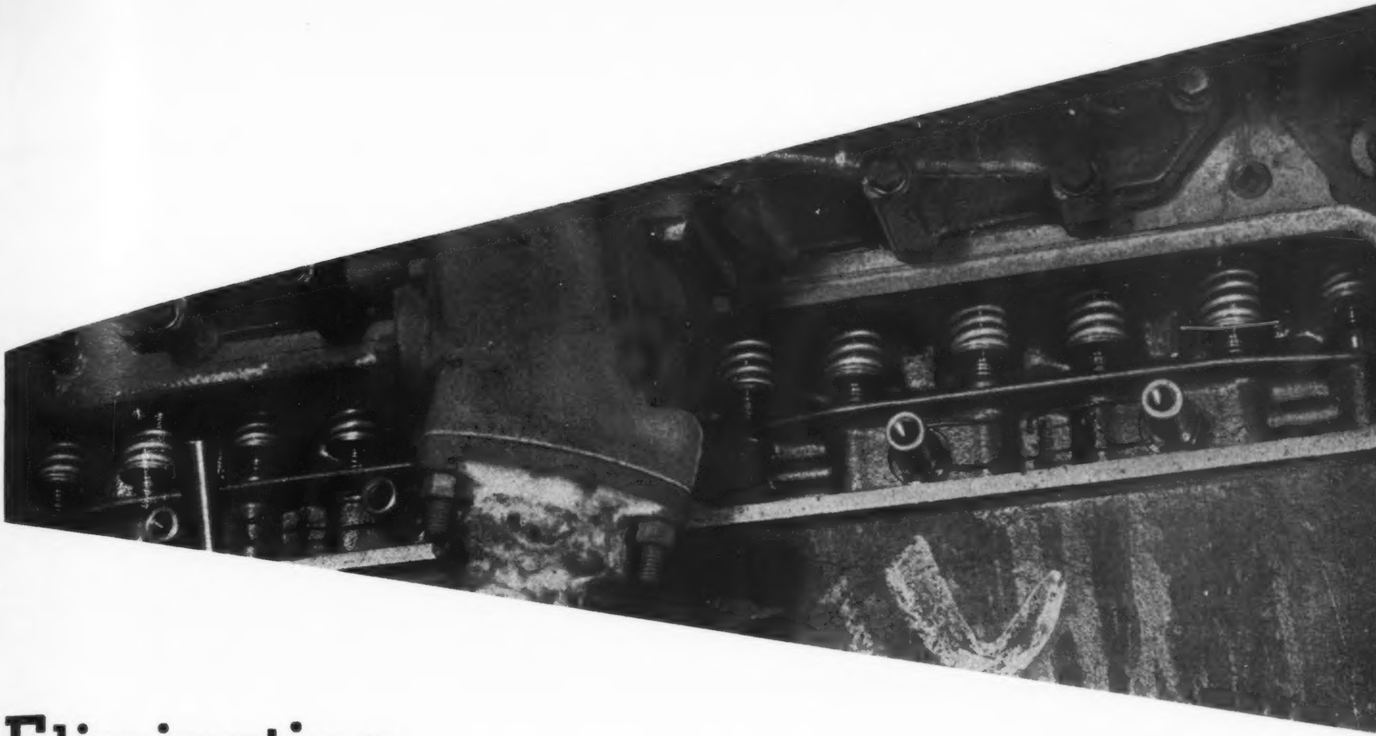
Fluorescent paint for aircraft dials, applied to the back of transparent materials and protected by a sealing coat of paint, gives more satisfactory service than when painted on the front where wear from fingers and abrasive materials cause deterioration. Dials shown at left are cellulose acetate butyrate and are made by the Erie Resistor Corp. Black paint backs the dials and is nonreflecting as is also the surface of the dials which is processed to obtain an etched face. An external source of ultraviolet radiations illuminates the symbols as shown.

Continuously varying temperature control, in accordance with a desired program, is simply effected by the Cenco thermostat at right through replacement of the manual control knob by a cam and lever arrangement. One revolution of the cam—hourly, daily or weekly—is a complete temperature cycle, the length of which is governed by the gearing system on the timer. Cams, cut to the shape of polar-type temperature charts, are easily made and readily changed for any desired program within the range of the thermostat. Possible applications include industrial ovens, certain researches and tests, incubators, etc.



Rotating cams interspaced between longitudinal bars, below, effect a gentle screening or sizing operation similar to that performed by inclined vibrating screens. Developed by the Roto-Flow Screen Co., action of the cams is sufficient to roll the material being screened from delivery end to discharge end. Cams are mounted on square shafts and may be reassembled for wider spacing if desired. Double chain and sprocket drive allows close center distances between camshafts. Action of screen is self-cleaning.





Eliminating Surging in Helical Springs

By A. M. Wahl
Westinghouse Research Laboratories

Fig. 1—Top—Surge-free operation is required of valve springs throughout wide speed ranges in internal combustion engines—Photo courtesy The White Motor Co.

Fig. 2—Above—Schematic of valve spring and gear

WHEN helical springs are subject to rapid reciprocating motion, additional stresses not present under static loading are involved. Under certain conditions this surging or vibration may be particularly severe and in extreme cases may lead to fracture or fatigue failure. The most important application where such vibrations must be provided for is the aircraft or automotive engine valve spring, *Fig.* 1. A sketch of a valve gear arrangement is illustrated in *Fig.* 2*a*, while a typical valve lift curve is shown in *Fig.* 2*b*. The latter represents the valve lift (or spring deflection beyond the initial value) plotted against time.

Another type of application of dynamically loaded springs is exemplified by the crank-type fatigue testing machine, *Fig.* 3, used for testing impulse turbine blades under alternating bending at high temperature[1]. The purpose of the springs in this case is to interpose flexibility between the crank and the test specimen so that an accurate alternating load on the latter can be maintained.

Springs Subject to Vibratory Loads: In the design of springs subject to a rapid reciprocating motion it is important to avoid, insofar as possible, a resonance between the frequency of the

[1] R. P. Kroon—"Turbine Blade Fatigue Testing", *Mechanical Engineering*, Vol. 62, Page 531 and Page 919.

alternating motion of the end of the spring and one of the natural frequencies of vibration of the spring. (Calculation of natural frequencies will be discussed later.) Therefore, if a spring is subject to reciprocating motion in a simple crank arrangement as indicated in *Fig.* 4 and provided the ratio r/l between crank radius r and connecting rod length l is not too large, the expression for the spring displacement from its position at top dead center is given with sufficient accuracy by the equation[2]

$$y=r+\frac{r^2}{4l}-r\left(\cos\,\omega t+\frac{r}{4l}\cos\,2\omega t\right)\qquad\qquad(1)$$

In this ω is the speed of the crank in radians per second ($\omega=\pi N/30$, where N is speed in revolutions per minute).

Two Frequencies Are Considered

This equation shows that for springs subject to this type of vibration there are two principal frequencies with which to be concerned: (1) The fundamental frequency of rotation represented by $\cos\,\omega t$ in Equation 1, and (2) a frequency twice this value represented by the last term in the equation. Thus, to avoid trouble from resonance, the spring should be sufficiently stiff that its lowest natural frequency, calculated from Equation 24, is considerably higher than twice the frequency of rotation of the crankshaft.

Where the spring is deflected by a cam as in valve springs, the valve lift curve $y=f(t)$ in *Fig.* 2*b* is not a simple sine function consisting primarily of one or two terms, but instead is a rather complicated function which may be assumed to consist of a large number of sinusoidal terms (a Fourier's series). Each of these terms may be thought of as acting separately. Thus the expression for valve lift (or spring deflection) becomes

$$y=f(t)=c_0+c_1\,\sin(\omega t+\phi_1)+c_2\,\sin(2\omega t+\phi_2)+\ldots\ldots+c_\mu\,\sin(\mu\omega t+\phi_\mu)+\ldots\ldots\qquad(2)$$

In this the fundamental frequency ω is equal to the camshaft speed in radians per second while the ϕ's represent the phase angles of the various harmonics. For practical purposes, the valve lift may thus be considered as a fundamental wave on which are superimposed various harmonics of higher frequency. In practice, harmonics as high as the twentieth may need consideration. In general, it should be noted that the amplitudes of these higher harmonics, represented by the terms c_1, c_2, etc., decrease as the order of the harmonic increases[3]. Usually it will be found difficult to avoid resonance within certain camshaft speed ranges between one of the higher harmonics and a natural frequency (usually the lowest) of the spring. When this takes place, vibration or surging of the spring occurs and this may increase the stress range by 50 per cent or more.

To reduce stresses resulting from such resonant vibrations in valve springs several methods are open. In the first place, the natural frequency of the spring may be made as high as possible so that resonance will occur only for the higher order harmonics of the valve lift curve, which are usually of lower amplitude. Hence, an improvement is obtained because the stresses set up by resonance with these higher harmonics are not as great as those set up by resonance with lower harmonics of greater amplitude.

Another method of reducing surge stresses in valve springs is to shape the cam contour so as to reduce the amplitudes of the harmonics which are of importance in the speed range within which the engine is to be used. For example, it might be found that for an engine with an operating range from 2000 to 3000 revolutions per minute, the 10th, 11th, and 12th harmonics are in resonance with the lowest natural frequency of the valve springs in this speed range. Hence a cam contour such as to give a low value for these harmonics would be of advantage. In many cases it is possible by a change in the cam contour to reduce the magnitudes of the harmonics to low values within certain speed ranges[4].

By reducing or varying the pitch of the coils near the ends of the spring, an improvement often may be obtained. The reason for this is that if resonance occurs with one harmonic these coils will close up, thus changing the natural frequency of the spring. This tends to throw it out of resonance[5]. Friction dampers, consisting of a three-

Fig. 4—Right—Diagram of spring subject to reciprocating action

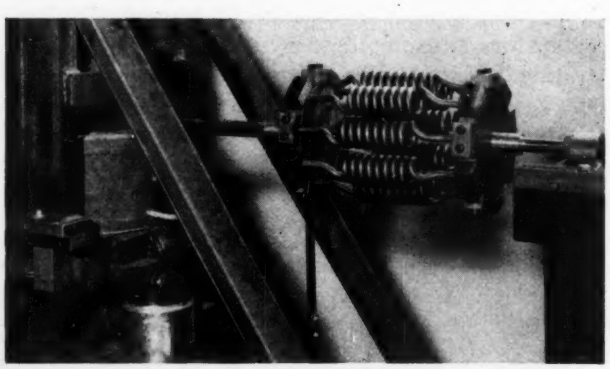


Fig. 3—Helical springs apply accurate loading on alternating bending machine

[2] See DenHartog—*Mechanical Vibrations*, Second Edition, Page 209, for derivation of this equation.

[3] The numerical values of the amplitudes of the various harmonics may be determined by harmonic analysis for any given valve lift curve.

[4] This is further discussed in *Schwingungen in Schraubenfoermigen Ventilfedern*—A. Hussmann, Dissertation, T. H. Berlin pub. by D. V. L., E. V. Berlin, 1938.

[5] *Die Federn*—Gross and Lehr, Page 115.

pronged device with the prongs pressing against the center coils of the spring, have also been used to damp out resonant oscillations[6].

DERIVATION OF EQUATION FOR VIBRATING SPRING: To calculate the natural frequency and vibration characteristics, it is first necessary to derive the differential equation of motion. Considering an element A of the helical spring in *Fig.* 5 which is compressed between the flat plates B and C, it is assumed that, when the spring is not vibrating, the element A, length of ds, is at a distance x from the left end of the spring. The active length of the compressed portion of the spring is taken as l while the effect of pitch angle will be neglected. If y is the deflection of the element from its mean position (or position when the spring is at rest) at any time t; if $\pi d^2/4$ is the cross-sectional area of the spring wire; if γ is the weight of the spring material per unit of volume and g the acceleration of gravity, then the mass of the element A is $\pi d^2\gamma ds/4g$. The force required to accelerate the element will be

$$F_a=\frac{\pi}{4}\frac{d^2\gamma dS}{g}\frac{\partial^2 y}{\partial t^2} \qquad (3)$$

This follows from the fundamental equation that force equals mass times acceleration since the acceleration of this element is $\partial^2 y/\partial t^2$. The partial derivative is used because y is a function of both s and t.

The change in y in a distance ds is $(\partial y/\partial s)ds$. For a complete turn this change becomes $\Delta y = 2\pi r\partial y/\partial s$ where r is the mean coil radius, and the total force P acting at a distance x will be, from the ordinary spring deflection equation

$$P=\frac{Gd^4\Delta y}{64r^3}=\frac{Gd^4}{64r^3}2\pi r\frac{\partial y}{\partial s} \qquad (4)$$

In this G is the modulus of rigidity.

Change in P in length ds is $(\partial P/\partial s)ds$ and is the net force F_b acting to accelerate the element A. Thus by differentiation of Equation 4, this force becomes

$$F_b=\frac{\pi}{32}\frac{Gd^4}{r^2}\frac{\partial^2 y}{\partial s^2}ds \qquad (5)$$

In addition there are damping forces from various causes including: (1) Internal hysteresis in the spring material, (2) air damping, (3) damping due to friction in the end turns and (4) damping from loss of energy in the supports. An exact method of taking all these into account would be hopelessly complicated. For mathematical convenience the damping force is assumed proportional to the velocity of motion. This means that if c is the damping force per unit length of the wire

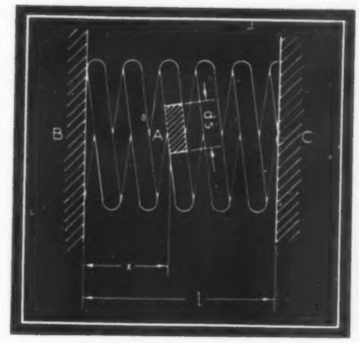


Fig. 5 — Helical spring compressed between parallel plates

per unit of velocity, the damping force F_d is

$$F_d=c\frac{\partial y}{\partial t}ds \qquad (6)$$

This force opposes the elastic force F_b. Hence from equilibrium

$$F_a=F_b-F_d \qquad (7)$$

or substituting Equations 3, 5 and 6 in Equation 7 and dividing by ds

$$\frac{\pi d^2\gamma}{4g}\frac{\partial^2 y}{\partial t^2}=\frac{\pi}{32}\frac{Gd^4}{r^2}\frac{\partial^2 y}{\partial s^2}-c\frac{\partial y}{\partial s} \qquad (8)$$

Since $s=2\pi rnx/l$ where l is the active length of spring and n the number of active turns,

$$\frac{\partial y}{\partial s}=\frac{l}{2\pi rn}\frac{\partial y}{\partial x}$$

and

$$\frac{\partial^2 y}{\partial s^2}=\frac{l^2}{4\pi^2 r^2 n}\frac{\partial^2 y}{\partial x^2}$$

by substitution of these values in Equation 8 and rearranging terms the following is derived:

$$\frac{\partial^2 y}{\partial t^2}+2b\frac{\partial y}{\partial t}=a^2\frac{\partial^2 y}{\partial x^2} \qquad (9)$$

where

$$W=\frac{\pi}{2}d^2nr\gamma$$
$$=\text{weight of active part of the spring} \qquad (10)$$

$$k=\frac{Gd^4}{64r^3n}=\text{spring constant} \qquad (11)$$

$$a=l\sqrt{\frac{kg}{W}} \qquad (12)$$

$$b=\frac{clg}{2W} \qquad (13)$$

In this the term b is a measure of the equivalent damping in the spring. In general b will vary with such factors as kind of material, amplitude of mo-

[6] "The Surging of Engine Valve Springs"—Swan and Savage, *Special Reprint* No. 10, Department of Scientific & Industrial Research, London.

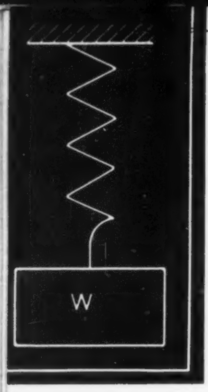


Fig. 6—Weight supported by helical spring

tion, design of end turns, rigidity of support, and can only be determined by actual tests on vibrating springs[7]. If the damping is zero, $b=0$ and Equation 9 reduces simply to

$$\frac{\partial^2 y}{\partial t^2}=a^2\frac{\partial^2 y}{\partial x^2} \qquad (14)$$

This is the same form as the known equation for longitudinal wave transmission in prismatical bars[8].

Natural Frequency of Spring: To calculate the natural frequency of the spring it is permissible to neglect damping since the small amount of damping present in actual springs does not affect the natural frequency appreciably. Hence the simpler differential expression in Equation 14 may be used. To solve this equation, it is assumed that

$$y=\phi(x)\times\psi(t) \qquad (15)$$

where $\phi(x)$ and $\psi(t)$ are functions of x and t, respectively. Then

$$\frac{\partial^2 y}{\partial t^2}=\phi\frac{d^2\psi}{dt^2}; \qquad \frac{\partial^2 y}{\partial x^2}=\psi\frac{d^2\phi}{dx^2}$$

Substituting these in Equation 14

$$\phi\frac{d^2\psi}{dt^2}=a^2\psi\frac{d^2\phi}{dx^2} \qquad (16)$$

or

$$\frac{1}{\psi}\frac{d^2\psi}{dt^2}=\frac{a^2}{\phi}\frac{d^2\phi}{dx^2}$$

This equation can only be satisfied if both members of the equation are equal to a constant, say $-\omega^2$. Then

$$\frac{d^2\psi}{dt^2}+\omega^2\psi=0 \qquad (17)$$

$$\frac{d^2\phi}{dx^2}+\frac{\omega^2}{a^2}\phi=0 \qquad (18)$$

Solutions of these equations are

$$\psi(t)=A_1 \sin \omega t+B_1 \cos \omega t \qquad (19)$$

$$\varphi(x)=A_2 \sin \frac{\omega x}{a}+B_2 \cos \frac{\omega x}{a} \qquad (20)$$

By substituting Equations 19 and 20 in Equation 15 a solution is obtained which satisfies Equation 14. The solution is

$$y=(A_1 \sin \omega t+B_1 \cos \omega t)\left(A_2 \sin\frac{\omega x}{a}+B_2 \cos\frac{\omega x}{a}\right) \qquad (21)$$

where A_1, A_2, B_1, B_2 are arbitrary constants depending on the boundary conditions.

Since both ends of the spring have been assumed as fixed or clamped, this means that regardless of the value of t

$$y=0 \text{ for } x=0$$
$$y=0 \text{ for } x=l$$

The first of these conditions requires that $B_2=0$, while the second requires that

$$\sin\frac{\omega l}{a}=0$$

To satisfy this equation requires that

$$\frac{\omega l}{a}=\pi,\ 2\pi,\ 3\pi,\ etc.$$

or

$$\omega=\frac{\pi a}{l},\quad \frac{2\pi a}{l},\quad \frac{3\pi a}{l},\ etc.$$

Since $\omega=2\pi f$ where f is a natural frequency of the spring, the natural frequencies are in the ratios 1:2:3, etc.

Using the value of a given by Equation 12, the expression for the natural frequency of the spring (in cycles per second) becomes

$$f=\frac{\omega}{2\pi}=\frac{m}{2}\sqrt{\frac{kg}{W}} \qquad (22)$$

where $m=1, 2, 3, \ldots$ is the order of the vibration.

Using the expressions for the spring constant k

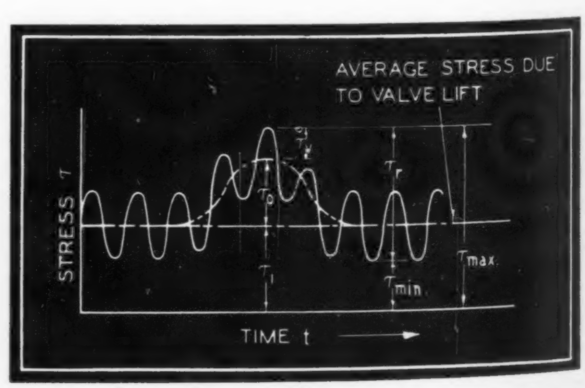


Fig. 7—Superposition of vibration stresses on stresses due to direct compression of valve spring

[7] See for example article by C. H. Kent, Machine Design, October, 1935, for a report of such tests. See also references of footnotes 4 and 6.

[8] Timoshenko—*Vibration Problems in Engineering*, Second Edition, Page 309.

and spring weight W given by Equations 11 and 10, the lowest natural frequency ($m=1$) becomes

$$f_1=\frac{d}{2\pi r^2 n}\sqrt{\frac{Gg}{32\gamma}} \qquad (23)$$

The lowest frequency is usually of most importance in practice.

This equation shows that, for a given material, the natural frequency of a helical spring is proportional to the wire diameter and inversely proportional to the product of the coil diameter and the number of active coils. For the usual steel springs where $G=11.5\times10^6$ lb./in.^{2} and $\gamma=.285$ lb./in.^{3} the formula for lowest natural frequency reduces to

$$f_1=\frac{3510\ d}{r^2 n}\ \text{(cycles per second)} \qquad (24)$$

For a spring with one end free and the other clamped, the lowest natural frequency would be equal to that of a similar spring twice as long but with both ends clamped. For such a spring Equation 24 may be used if the number of turns is taken as twice the actual number in the spring.

As an example of the use of these equations in calculating natural frequency, assuming a steel spring with $d=.3$-inch, $r=1$ inch, $n=6$, Equation 24 becomes

$$f_1=\frac{3510\times.3}{(1)^2\times6}=175\ \text{cycles per second}$$

In the second mode of vibration the spring frequency will be double this, or 350 cycles per second.

For a spring with a weight hanging on its end as shown in *Fig.* 6 the lowest natural frequency of the system may be calculated as follows: It is known that where a mass is deflected by a certain amount δ under its own weight (the mass of the spring being small compared to that of the weight) the natural frequency (in cycles per second) may be taken as [9]

$$f=3.14\sqrt{\frac{1}{\delta}} \qquad (25)$$

In the case where the mass is supported by a helical spring of appreciable weight as indicated in *Fig.* 6, it has been found that if the weight of one-third of the spring is added to that of the mass W_m, the calculated deflection may be used for figuring the lowest natural frequency. If W is the spring weight from Equation 10 and k the spring constant in pounds per inch deflection, the frequency in cycles per second becomes

$$f=3.14\sqrt{\frac{k}{W_m+\frac{1}{3}W}} \qquad (26)$$

[9] *Mechanical Vibrations*, Page 45.

Surging of Engine Valve Springs: Since most aircraft and automobile engines run at variable speeds as mentioned previously, it is practically impossible to avoid resonance between one of the higher harmonics of the valve lift curve and a natural frequency of the spring at some speed of operation. When this occurs, the amplitude of vibration and the resulting stress in the spring depend primarily on the amplitude of the harmonic which is in resonance and on the amount of damping in the spring, represented by the damping term b in Equation 9.

To obtain the additional stress in the spring due to this vibration, it is assumed that one end of the

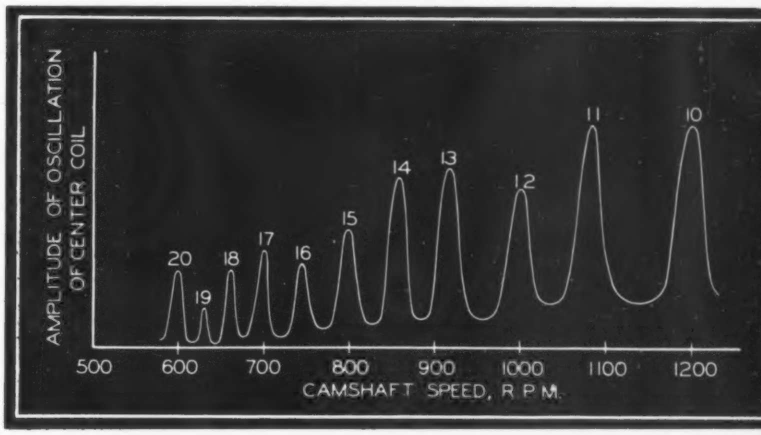


Fig. 8—Typical shape of resonance curve of valve spring. Order of harmonic is noted at each resonance peak

spring in *Fig.* 5, for instance end *C*, is oscillated through an amplitude represented by the function

$$y_o=c_n \sin \omega_n t$$

In this c_n represents the amplitude of the particular harmonic of the valve lift curve which is in resonance with a natural frequency f_n of the spring, usually the lowest. The circular frequency of this harmonic is taken as $\omega_n=2\pi f_n$. The amplitude of motion and stress due to this harmonic may be determined by solving Equation 9 in conjunction with the proper boundary conditions. This stress is then superimposed on the static stress due to the valve lift as indicated in *Fig.* 7. The dot-dash line represents the stress due to the valve lift, the maximum value being τ_o. On this is superimposed a higher frequency vibration represented by the stress τ_v which is due to resonance with a given harmonic of the valve lift curve.

To solve Equation 9 for the *steady state* condition, the deflection y at any point x from the end of the spring is assumed given by

$$y=F(x)\ \sin\ \omega_n t \qquad (27)$$

where $F(x)$ is a function of x only. This, of course,

(*Continued on Page* 216)

Mobilizing the Graduate into War Work

By Colin Carmichael

IN the present all-out war effort the transition from student engineer to practicing engineer is unprecedentedly abrupt. Analyzing the young engineer's background in relation to his start in industry, the author submits pertinent suggestions which should be invaluable in accelerating the effective utilization of the students' potentialities and technical ability

COLLEGE graduates are raw materials that must be processed to fit the specific needs of the jobs they are intended to fill. Training programs, ranging in length from six months to three years, are sponsored by many concerns. Other companies admit, with some complacency, that college graduates do not begin to earn their pay for about two years after graduation. This time lag is a challenge to chief engineers and college graduates alike. Is it possible that failure of college men and professional engineers to understand each other is responsible? The views of both groups point to the fact that misunderstandings play a large part in slowness of adjustment.

Initial dissatisfaction with the newly-employed graduate, according to a recent survey[1], is often due to personal traits. Further, young engineers are found to be over-eager for advancement, unwilling to prepare for advancement over a long period of time, lacking in aptitude for engineering work, unable to do routine work and at the same time avoid getting into a rut, lacking in initiative and lacking in self-confidence. To this list other comments might be added. It is always easy to criticize, but not so easy to be constructive. Engineers need not, however, be content with talking about the slow adjustment of college graduates to their jobs, they can do something to help speed it up. To this end, the author has drawn upon the experience of eleven years on the assembly lines of college graduate production, to present a picture of the young engineer and his background and to suggest means whereby his better qualities may be developed and his traits overcome.

Many engineers find college graduates at first unable to get along with their associates. Elimination of this difficulty would do more to speed up adjustment of the young engineer to his new environment than any other single factor. With few exceptions, college students are highly gregarious. They co-operate with each other in various enterprises, including study, and should be as fully capable of adapting themselves to the peculiarities of others as any other group. Furthermore, they understand the value of team spirit in working toward a common goal.

Impeded by Personal Relations

Can the acknowledged failure of this attitude when he enters industry be blamed entirely on the young graduate? It is an unfortunate consequence of human nature that a mutual attitude of distrust often exists between the graduate with formal education and the man with experience. In some cases the experienced man lacks a college education, and resents the advantage that the young man has gained. His resentment may show in a refusal to help the youngster to become conversant with his new job, or it may even develop into active hostility.

A superior attitude on the part of the graduate is sometimes present, based perhaps on social background as well as on a feeling that his degree confers certain "rights and privileges" denied to others. Another factor that will influence this

[1] *The Journal of Engineering Education,* May, 1938, Page 598.

year's graduate, in particular, is the surprising number of offers of jobs that he will have received. It will be natural for him to feel that he is a person of considerable importance, sought after by the best engineering concerns in the country, and fully conscious of the favor he is conferring on the company of his choice. This attitude is not calculated to endear the young man to the more experienced men with whom he must work. Engineers may save considerable lost motion in their departments if they keep on the lookout for this attitude. A private talk, coming from someone in authority, will be heeded. The student has often listened to advice from deans, professors and faculty advisers, and has come to recognize when advice is being offered for his own good.

The long hard road of adjustment may be better as a character molder, but time is short. The young graduate, being responsive to the team spirit, can be counted upon in the present war effort to forget personal problems. The engineer in charge, for his part, should assume some responsibility as a morale builder.

Background of Designer

An important factor in the attitude of the student toward design work is to be found in the methods of teaching the basic design courses. Technical course sequences that lead directly to mechanical design are shown in *Fig.* 2. Most of the material in the prerequisite courses will be used directly by the designer, so that the total design sequence may be said to approximate two-thirds of the mechanical engineering curriculum. The remaining one-third comprises the other mechanical engineering courses (heat power, aerodynamics, etc.), civil and electrical engineering service courses, and electives, large portions of which are also of direct concern to the designer. In the middle block of *Fig.* 2, Mechanism and Machine Design (7 per cent) might be replaced with Heat Power (13 per cent), Aerodynamics (3 per cent) or some other branch of mechanical engineering, and with slight shifts in emphasis, the chart becomes a heat power sequence, or an aerodynamic sequence, etc. Thus there is keen competition at the focal point for the students' attention. It is in these application courses that the professional point of view comes to the fore.

Few teachers of machine design have themselves done practical design. One result is that in many schools machine design courses lack the breath of life necessary to stimulate the students' interest. The creative aspects of design are neglected, and the highly important qualities of ingenuity and originality are likely to be subordinated to analytical ability in measuring the student's success in his machine design courses. Consequently, misconceptions arise regarding the nature of design work. To many students design consists of "pushing a pencil across a drawing board" or "sitting in an office working problems".

Origin of the widespread aversion to drafting board work is usually the freshman mechanical drawing course in college. The exercises the student worked out were often meaningless chores which he was unable to connect with reality. The class was usually a large one and the discipline strict. At the same time, the fact that his drafting experience occurred so early in his college career gives him the idea that it is something elementary, since most of his later engineering courses required no drafting.

Ignorance of the true nature of design work is

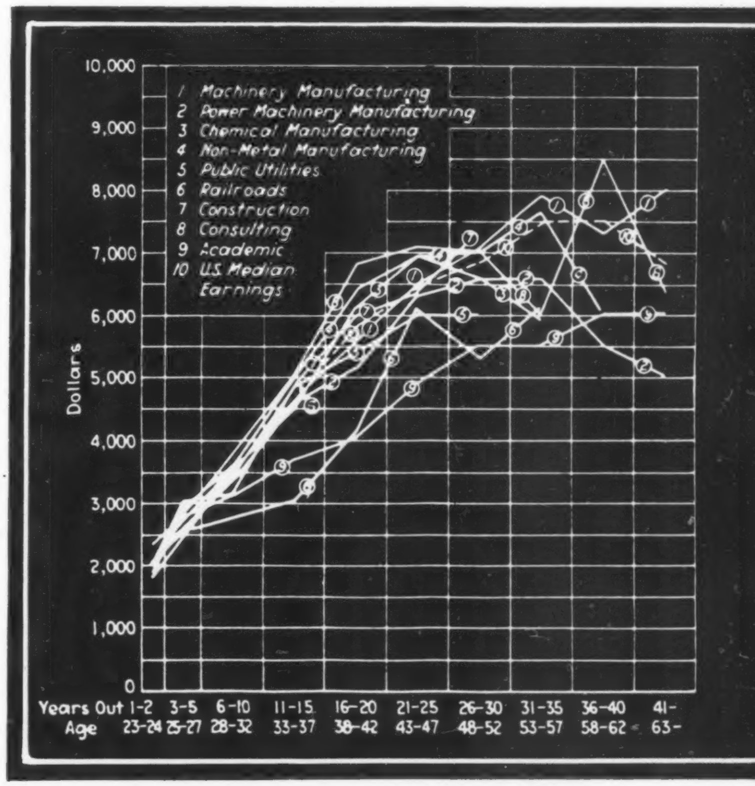


Fig. 1—Trend of earnings in machinery manufacturing companies is appreciably above the median for the engineering industries of the United States

quite prevalent, according to J. H. Belknap of the Westinghouse company in a recent paper "Why do graduating engineers frequently avoid positions in the design department?"[2]

Chief engineers with the opportunity to pick young college men for their departments from among those hired by the personnel office will do well to ignore any preferences expressed by the youngsters for production, sales, research, and so forth. Graduates should be given the opportunity to talk with the better men with the idea of attracting them to design work. Aptitudes and interests which were hidden because of the con-

[1]Presented at a conference on mechanical enginering instruction, S.P.E.E.-A.S.M.E., Purdue university, June 30, 1941.

ditions just outlined may be brought out if the engineer can get across to the boys some of his own enthusiasm for his job.

It will astonish the boys to learn how little time the chief designer spends at his desk compared with that which he spends in conferences with production and sales engineers, in following up the manufacture and testing of his projects, and in other activities which are not in the least "confining." The important difference between design, which is creative, and drafting, which is interpretative, should be stressed. At the same time it might be pointed out that a start in the drafting room should be considered a privilege, because of the unique experience it affords the young engineer of seeing the ideas conceived in the minds of top-flight designers take practical shape on the drawing board. Finally, the engineer should try to give the prospective designer an insight into the peculiar joys and satisfactions of design work, of seeing one's own ideas take shape.

Engineers who come in contact with or supervise the work of a young graduate can help him fill some of the gaps that necessarily occur in his earlier training by directing his attention to the books and periodical literature relating to his

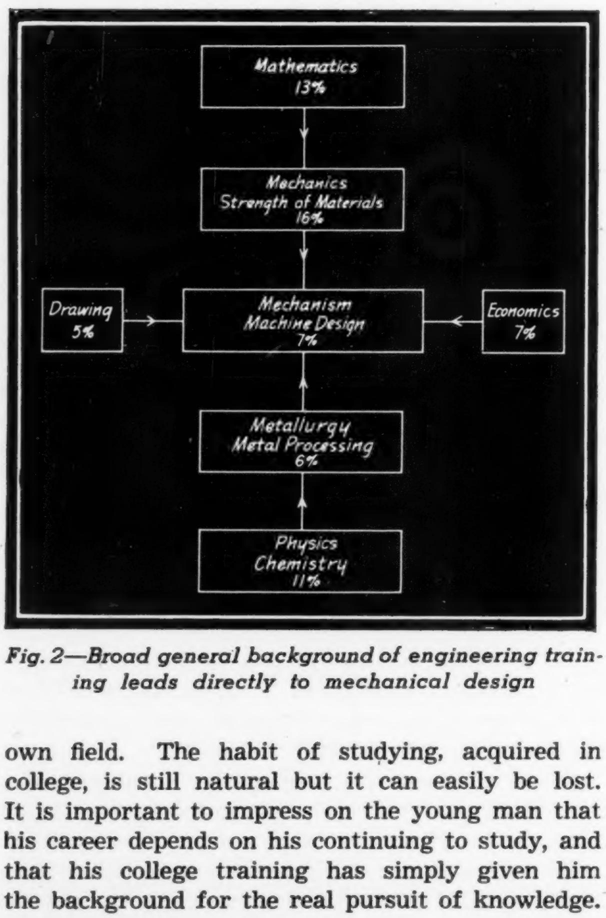


Fig. 2—Broad general background of engineering training leads directly to mechanical design

own field. The habit of studying, acquired in college, is still natural but it can easily be lost. It is important to impress on the young man that his career depends on his continuing to study, and that his college training has simply given him the background for the real pursuit of knowledge.

Basic training which the young graduate has received covers a wide field and is up-to-date. Chief engineers will find it to their advantage to draw upon the stock of knowledge which the recent graduate has acquired in other fields. The young man's enthusiasm will be aroused by the feeling that he can contribute something constructive early in his career; too often he is only aware of the things he doesnt know and can't do—yet.

Practical designers are prone to ridicule the "theoretical" designers, after whom the graduates fresh from college are patterned. They may be mathematical wizards, but when it comes to establishing the dimensions of an ordinary machine part they are at a complete loss or, worse still, they settle on absurd dimensions, ignoring practical limitations. The problems on which the student works in college are mostly intended to bring out the application of some particular theory.

For example, in stress analysis, when he has established a shaft dimension that gives a safe working stress, he hands in the problem, receives a satisfactory grade, and goes on his way convinced that he is a competent designer of shafting. This recitation did not happen to cover shaft deflections, transverse or torsional vibrations, or bearing design, and there was not time in class, after the students had been drilled in combined bending and torsion theory, to discuss practical design aspects.

The student's experience in establishing dimensions of machine parts is thus almost wholly confined to those that can be calculated. Even in courses specifically devoted to machine design this is often true, though teachers who have themselves done some design try to give practical considerations their proper weight and give the student some chance to exercise his judgment.

Development of Judgment Important

Judgment, however, is a quality sadly underdeveloped in college. For the instructor with a heavy teaching load and many papers to grade the temptation to give all the data needed to work a problem, so that there is only one correct answer, is well-nigh overwhelming. The result is that the student rarely has to assume anything or look up any information in sources outside his course textbook. Engineers are familiar with the result of this defect in college training, and if they also understand the cause of it they may be more sympathetic towards the initial difficulties of the young graduate and, without belittling the value of theoretical training, seek to impress upon him the importance of developing his judgment and initiative.

Experience, may it be repeated, is often the best teacher but it is certainly not the fastest. Particularly in our present all-out war effort, time is at a premium. Responsibility, therefore, rests equally upon the shoulders of the chief engineer and the graduate that the period of adjustment of the latter may be reduced to a minimum and the day hastened when he can assume his proper portion of the creative work which ultimately will lead to victory.

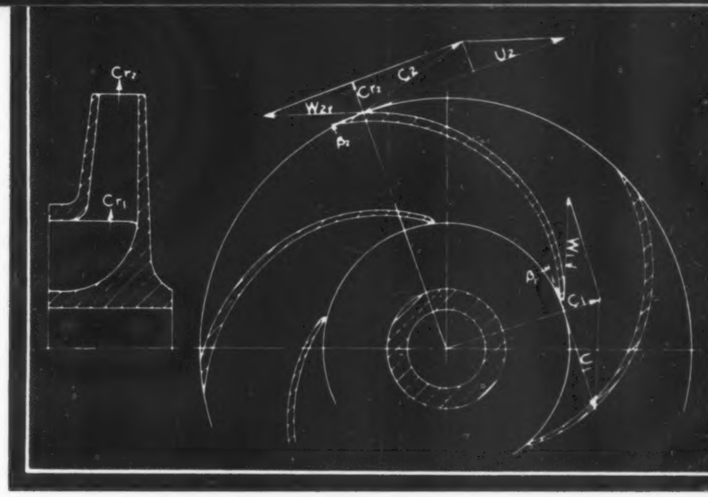


Fig. 1—Section through typical impeller shows diagrams of velocities at inlet vane tips and periphery

Pump Design Affords Wide Scope in Application

By H. Gartmann

De Laval Steam Turbine Co.

UNDERSTANDING of the limitations and potentialities of centrifugal-type fluid pumps is essential to their proper application. A fundamental analysis of pump characteristics and the design features which affect them afford perhaps the only practical basis for such an understanding. Beginning with a development of underlying theory, this article discusses the manner in which special operation may be attained with substantially standard pumps.

In order to deduce a formula for the head theoretically developed by a pump impeller, it is assumed that the impeller has an infinite number of vanes and is handling a frictionless fluid. The head developed by the impeller is the total energy of one pound of liquid at the outlet of the impeller minus its total energy at the inlet. Increase in energy comprises, first, increase in static pressure, mostly due to centrifugal force of the rotating liquid and, second, increase in velocity head corresponding to the increase in absolute velocity between inlet and outlet of the impeller. *Figs.* 1 and 2 show sectional views of a pump impeller and the velocity diagrams for the impeller inlet and the impeller outlet. In the diagrams,

C_1 and C_2 are the absolute velocities of the liquid at the impeller inlet and outlet, respectively, in feet per second

U_1 and U_2 are the peripheral velocities of the impeller tips at inlet and outlet, in feet per second

W_1 and W_2 are the velocities of the liquid relative to the impeller at the points of entering and leaving the vanes, respectively, in feet per second.

At no flow, the increase in static head energy resulting from the centrifugal action of the rotating liquid inside the impeller is $(U_2^2-U_1^2)/2g$ feet, where g is the acceleration of gravity in feet per second per second. Actually there exists inside the impeller a relative fluid velocity changing from W_1 at the inlet tip to W_2 at the outlet. As W_2 is usually less than W_1, the static head at the outlet is increased by the difference in velocity head, or $(W_1^2-W_2^2)/2g$, and the static head at the outlet accordingly is

$$\frac{U_2^2-U_1^2}{2g}+\frac{W_1^2-W_2^2}{2g}$$

The energy increase due to the increase in abso-

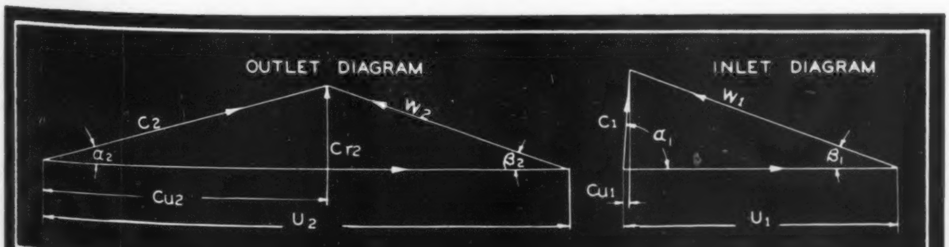


Fig. 2 — Simplified inlet and outlet diagrams for impeller shown in Fig. 1

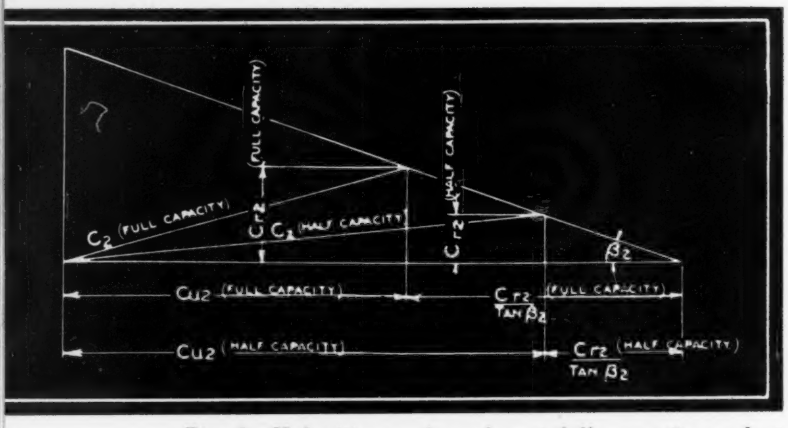


Fig. 3—Velocities and angles at full capacity and at half capacity for impeller of Fig. 1

lute velocity between inlet and outlet of the impeller is $(C_2^2—C_1^2)/2g$, and adding this to the static energy increase, gives for the total energy increase, expressed in head of liquid

$$H=\frac{(U_2^2-U_1^2)+(W_1^2-W_2^2)+(C_2^2-C_1^2)}{2g}$$

Referring to *Fig.* 2, W_1 and W_2 can be expressed as functions of U, C and the absolute angle α between U and C, and the formula becomes

$$H=\frac{1}{g}(U_2C_2\cos\alpha_2-U_1C_1\cos\alpha_1)$$

or, since $C_2cos\alpha_2$ and $C_1cos\alpha_1$ are the components of C_2 and C_1, respectively, in the direction of U

$$H=\frac{1}{g}(U_2Cu_2-U_1Cu_1) \quad\text{..........................}(1)$$

In a pump without inlet guide vanes, the liquid enters the eye of the impeller in a nearly axial direction and, after turning 90 degrees, enters the impeller at the inlet vane tips in an almost radial direction. Therefore, the inlet velocity component Cu_1 is practically zero and the head formula simplifies still further to

$$H=\frac{1}{g}U_2Cu_2 \quad\text{..................................}(2)$$

Head actually developed by a pump is substantially less than the foregoing theoretical considerations would indicate, due to various causes to be enumerated later. However, by assuming that Equation 2 represents the head developed by a pump, certain conclusions regarding the characteristic curve can be drawn.

With constant speed of rotation, U_2 is constant. Component Cu_2 equals $U_2—Cr_2/tan\beta_2$, where Cr_2 is the radial outlet velocity at the impeller periphery and β_2 is the outlet vane angle. Substituting this term in the second member of Equation 2

$$H=\frac{1}{g}U_2\left(U_2-\frac{Cr_2}{\tan\beta_2}\right)$$
$$=\frac{1}{g}U_2^2\left(1-\frac{Cr_2}{U_2\tan\beta_2}\right) \quad\text{.....................}(3)$$

When the capacity of a given pump, operating at constant speed, changes, only Cr_2 in the above formula will change, and as the outlet area of the wheel is constant, Cr_2 will change in direct proportion to the delivery, *Fig.* 3, and the theoretical head-capacity curve becomes a straight line.

Further, for outlet vane angles less than 90 degrees, that is, for backwardly curved vanes, the head decreases with increasing delivery since Cr_2 increases in the same ratio. For a vane angle of 90 degrees, $tan\ \beta_2$ is infinity, that is, the head is constant for all deliveries. For vane angles larger than 90 degrees, $tan\ \beta_2$ is negative, making $Cr_2/U_2\ tan\ \beta_2$ positive and the head increases with increasing delivery, *Fig.* 4. It will be seen that for no delivery or shut-off, where $Cr_2/tan\beta_2$ is zero for any angle, the theoretical head becomes U_2^2/g.

Actual Differs from Theoretical

Actual characteristic curve of a centrifugal pump differs considerably from the theoretical head-delivery curve shown in *Fig.* 4 because of the fact that the water does not leave the impeller blades tangentially, but at an angle always smaller than the vane angle, and also because of friction and turbulence losses. However, it still retains the essential characteristics here developed for a theoretical impeller.

Figs. 5 and 6 illustrate the relations graphically.

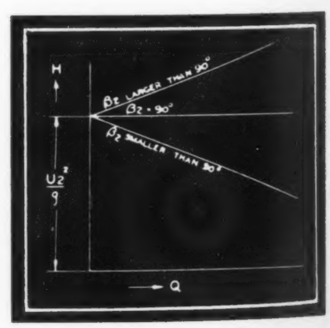


Fig. 4—Theoretical head-delivery curves for various values of the vane outlet angle

As has been shown, the principal factors influencing the shape of the characteristic curve are the angles of the impeller vanes at their delivery ends or tips and the magnitude of the radial velocity. If the vanes are curved backward, as in standard practice today, the head will fall off as the delivery increases. By suitable design of the vane angles and the outlet area of the wheel, the head may be made to fall off sharply.

Where the pump is to work against practically

constant head at variable delivery, as for instance in boiler feed service, a flat characteristic is desirable, while for variable head and nearly constant delivery a steep characteristic curve is more suitable. In either case, the curve should steadily decrease in head, from shut-off pressure to maximum delivery, especially if two or more pumps are to operate in parallel. If, in the case of two pumps operating in parallel, the shut-off pressure were less than the head at some other point on the curve and the pump in service were operating at the high point, it would be impossible to get the second unit on the line without increasing its speed or resorting to the use of a circulating connection or by-pass so that the pump could be operated at the required head before being placed in parallel operation.

Outlet Velocity Is High

Equation 3 also shows why high specific speed pumps, such as mixed-flow pumps, have inherently steep characteristic curves. With the small diameter wheel, fairly small outlet angles must be used in order to get vanes of sufficient length to guide the fluid properly. Also, the radial outlet velocity, due to the large capacities handled and the limitation in wheel diameter and available outlet area, is comparatively high, giving a large value of $Cr_2/U_2 tan\beta_2$, which obviously results in a steep characteristic.

Thus far, only the influence of the impeller on the shape of the characteristic curve has been discussed, no reference having been made to the influence of the surrounding casing. As shown in developing Equation 1, the total energy of the liquid at the discharge of the impeller comprises two quantities. The static head at this point is composed of the sum of these quantities and is represented by the expression

$$\frac{(U_2^2-U_1^2)}{2g}+\frac{(W_1^2-W_2^2)}{2g}$$

while velocity head, or kinetic energy, is represented by the expression $(C_2^2—C_1^2)/2g$ or, as the magnitude of $C_1^2/2g$ is usually negligible, by $C_2^2/2g$. The ratio of these two factors varies according to the angle of the impeller vanes and also according to the ratio between the velocity W_2 of the water relative to the impeller and the peripheral velocity U_2 of the impeller. As can be seen from the outlet diagram of *Fig.* 2, the kinetic energy, or velocity head, represented by $C_2^2/2g$ is in general large for radial vanes and diminishes with decrease in outlet angle β_2.

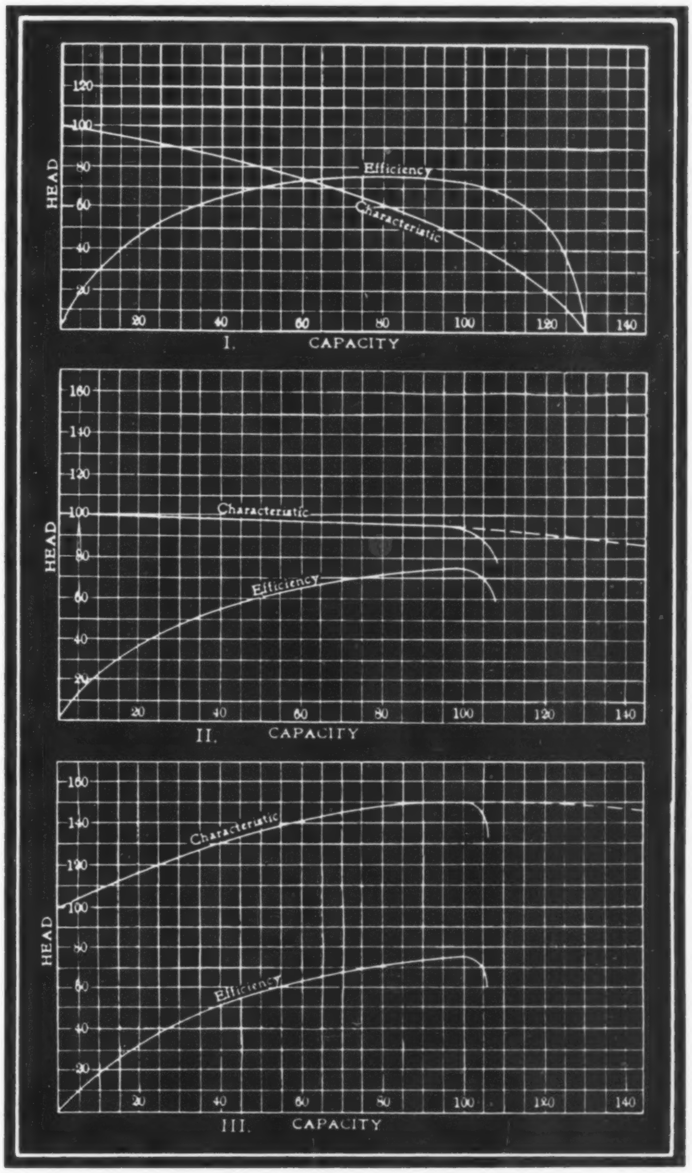


Fig. 6—Characteristic curves corresponding to impeller vane shapes shown in Fig. 5

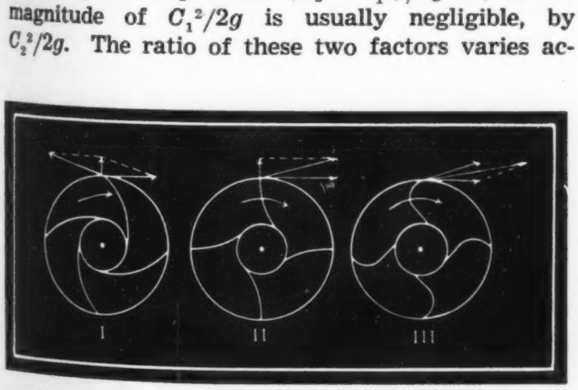


Fig. 5—Three typical shapes of impeller vanes

Efficiency of the pump as a whole depends, first, upon the efficiency with which the impeller pressure is generated, and second, upon the efficiency with which the velocity head $C_2^2/2g$ possessed by the water as it leaves the impeller is converted into pressure head. This will also affect the final characteristic curve. Efficiency of conversion within the impeller is increased by correct surfaces, gradual change of areas, and smooth finish. It is advantageous to develop a large part of the total head in the impeller, where velocities

are relatively low and where the conversion of energy into pressure can be effected efficiently. Moreover, the existence of a large portion of the energy as velocity head in the water as it leaves the impeller is accompanied by an unsuitable form of head-delivery curve.

Two principal methods are employed for the conversion of velocity head. With low and medium head pumps, a spiral casing or volute is used in which conversion takes place according to the law of conservation of moment of momentum, namely, that the velocity of each particle of the water in a circumferential direction will vary in the inverse ratio of its distance from the center. In other words, the larger the ratio between housing and wheel diameter, the more the velocity will be reduced and the more the kinetic energy of the water as it leaves the impeller will be converted into pressure by the time the water has reached the periphery of the diffuser space. The relatively small impellers employed in pumps of high specific speed lend themselves admirably to this method of diffusion, since the ratio of the outer radius of the diffusing chamber to the inner radius can be made very large without unduly increasing the size of the pump casing.

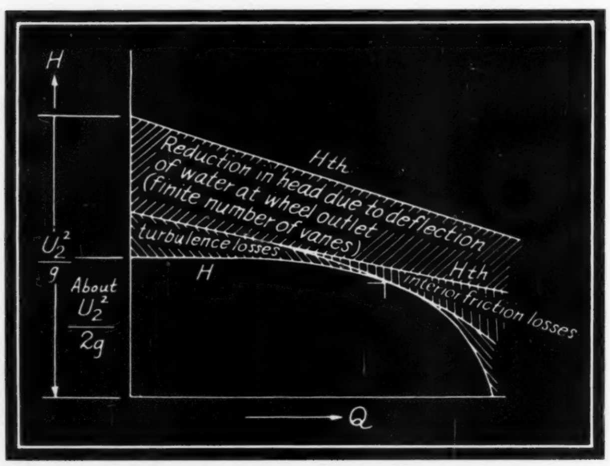


Fig. 7—Development of actual head-delivery or test curve H from the theoretical head-delivery curve

For high head pumps, particularly multistage pumps, diffusing vanes are usually employed for converting the velocity head of the water as it leaves the impeller into pressure head. By employing a number of vanes placed around the periphery of the impeller so that they form passages of gradually increasing cross section, the high velocity of the water as it leaves the impeller is efficiently converted into pressure in a relatively short radial distance.

The head-capacity curve which has been developed herein, theoretically is a straight line. For outlet angles of less than 90 degrees it has the shape shown in *Fig.* 7 for *Hth*. However, the water in an actual pump impeller with a finite number of vanes will always leave the impeller at an angle β_2 relative to the tangent at the periphery which is smaller than the actual vane angle. This reduces the theoretical head developed in a practically constant ratio, even if a frictionless fluid is considered, and the new theoretical head has the shape of Hth_1 in *Fig.* 7. The reduction in head from Hth to Hth_1 does not represent loss in pump efficiency, but takes into account discrepancies between the theoretical flow of a frictionless fluid in an impeller having an infinite number of vanes and the actual flow in an impeller having only a few vanes. The friction losses in the suction passages, inside the wheel and in the discharge volute will be least at no discharge and will increase approximately as the square of the delivery, reducing the head Hth_1 approximately as shown.

Losses Caused by Turbulence

Losses, next to be considered, are caused by turbulence. Referring to the inlet diagram of *Fig.* 2, it will be seen that for a given wheel operating at a fixed speed, that is, with U_1 constant and inlet angle β_1 fixed, there is only one inlet velocity C_1, that corresponding to the rated capacity, which will fulfill the conditions of the diagram. For reduced or increased delivery the water enters the wheel at an angle different from the vane angle, with resultant shock and turbulence losses. Turbulence losses also occur at the entrance to the diffuser in diffusing vane pumps and, to a smaller extent, at the tongue of a volute pump, at deliveries different from rated capacity. The final head-delivery curve H accordingly appears as in *Fig.* 7. It is difficult to build a diffusing vane pump with a head curve steadily decreasing from shut-off to full capacity. The turbulence losses in diffusing vane pumps at reduced deliveries are considerable since the angle of the diffuser vane is correct for only one capacity. However, during recent years this disadvantage has been largely overcome by specially dimensioning the diffuser vanes so that this type of pump can be built with a stable characteristic.

For most pumps the head at no delivery or shut-off pressure has a value of about $U_2^2/2g$. (The actual value varies from about $.9U_2^2/2g$ to $1.1U_2^2/2g$). Pressure at shut-off is then

$$H=\left(\frac{N\times D}{1840}\right)^2$$

wherein N is revolutions per minute and D, wheel diameter in inches. If there is no flow, the static pressure developed in an impeller is $(V_2^2—V_1^2)/2g$, or if all the water down to the wheel hub is rotating, very nearly $V_2^2/2g$. Even though the conditions under which an impeller operates at zero delivery vary considerably from the theoretical assumptions, the head developed is actually very nearly $V_2^2/2g$, and this formula can conveniently be used for calculating the approximate impeller di-

(*Concluded on Page* 214)

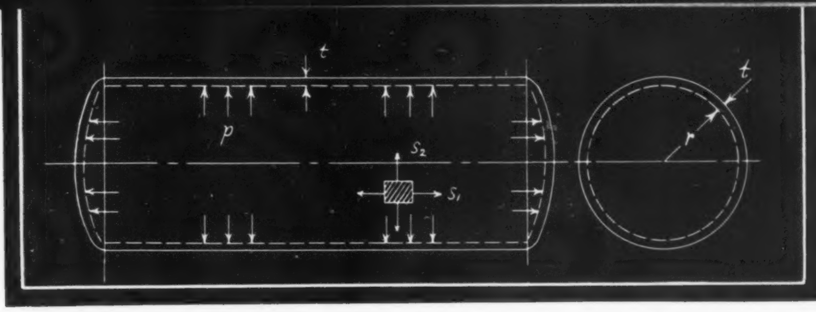


Fig. 1—Stresses in element of cylindrical vessel with internal pressure

Determining Wall Thickness for Pressure Vessels

By Joseph Marin

Pennsylvania State College

DESIGN of equipment utilizing pressure vessels, storage tanks and piping involves the determination of wall thickness required. This article presents an answer to the problem, using the distortion energy theory of failure to allow for the effects of a combined state of stress. For convenience, design diagrams are included to determine the wall thickness required for a specific application.

Problems treated will be confined to thin-walled constructions. With this limitation, the stresses in the tanks and pressure vessels considered are assumed free from bending. These stresses, called membrane stresses, do not vary across the wall provided the thickness of the wall is small.

Also discussed in the article is the design of pressure piping subjected to bending, twisting, axial loading and internal pressure. Only thin-walled piping for relatively low pressures will be considered. This type of piping, however, is encountered in many applications.

PRESSURE VESSELS: Many containers are used for the storage of liquids and gases in which an internal pressure is produced. These containers, or pressure vessels, usually are cylindrical with hemispherical, ellipsoidal or conical heads[1]. In determining the wall thickness, numerous factors must be considered in addition to the effect of internal pressure. Some of these are the influence of impact loads caused by rapidly fluctuating pressures, weight of the vessel and normal contents at operating conditions, superimposed loads, wind loads, localized stresses due to supporting lugs or saddles, bending stresses at junction of shell and the heads at the ends of the vessel, and temperature effects. Allowances also must be made for corrosion and for the efficiency of the riveted or welded joint. A discussion of these factors is given in the *Code for Unfired Pressure Vessels for Petroleum and Gases*[1].

Determination of wall thickness required in vessels considering the pressure only is usually based on a consideration of the maximum stress. In the following, the thickness required for both the shell and various types of heads will be determined by considering the effect of the combined state of stress present. This will be done by basing the analysis on the distortion energy theory.

Example 1—Cylindrical Pressure Vessel: For the cylindrical part of the pressure vessel in *Fig.* 1 subjected to an internal pressure p, the stresses S_1 and S_2 produced on the element shown are

$$S_1=\frac{pr}{2t} \quad \text{and} \quad S_2=\frac{pr}{t} \qquad (a)$$

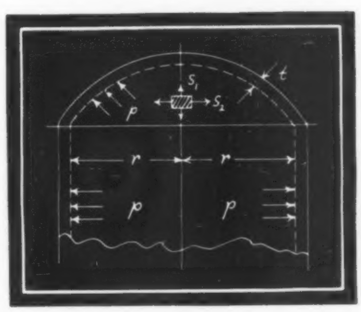


Fig. 2—Membrane stresses in cylinder head element due to force p

where r is the inside radius and t the wall thickness.

The distortion energy for this element is

$$V=c(S_1^2-S_1S_2+S_2^2) \qquad (1)$$

or

[1] See, for example, A.P.I.—A.S.M.E. *Code for Unfired Pressure Vessels for Petroleum Liquids and Gases*, Third Edition, 1938.

$$V = .75\left(\frac{pr}{t}\right)^2 \qquad (b)$$

Equating this to the allowable value cS_w^2, where S_w is the working stress, and solving for t, the required wall thickness is

$$t = .86\frac{pr}{S_w} \qquad (2)$$

By the usual stress theory the thickness is obtained by equating the maximum of the stress, as in Equation *a*, to S_w or

$$t = \frac{pr}{S_w} \qquad (2a)$$

This thickness is 14 per cent greater than that given by Equation 2.

Example 2—Cylinder Head—General: Membrane stresses due to pressure produced in the cylinder head of the vessel, *Fig.* 1, will depend on the shape of this shell. The stresses have been obtained for a vessel represented by a general surface of revolution[2], as shown in *Fig.* 2. Stresses S_1 and S_2 are

$$S_1 = \frac{pR_2}{2t} \quad \text{and} \quad S_2 = \frac{pR_2}{2t}\left(2 - \frac{R_2}{R_1}\right) \qquad (c)$$

where R_1 is the radius of curvature of the meridianal section and R_2, the radius of curvature of the section perpendicular to the meridian at the point considered. Using Equation 1, the distortion energy for the element is

$$V = c\left(\frac{pR_2}{2t}\right)^2\left[3 - 3\frac{R_2}{R_1} + \left(\frac{R_2}{R_1}\right)^2\right] \qquad (d)$$

Equating this energy to the allowable value cS_w^2, the required wall thickness t in terms of the radii of curvature R_1 and R_2 is

$$t = \frac{pR_2}{2S_w}\sqrt{3 - 3\frac{R_2}{R_1} + \left(\frac{R_2}{R_1}\right)^2} \qquad (3)$$

Fig. 3 represents the variation in this wall thickness as a function of the ratios R_2/R_1 and R_1/R_2 with values from 0 to 1. In this way all possible shapes of shells are considered.

Example 3—Hemispherical Head: For a hemispherical shaped head, $R_2 = R_1$ in Equation 3 and the wall thickness is

$$t = \frac{pr}{2S_w} \qquad (4)$$

This agrees with the wall thickness as usually de-

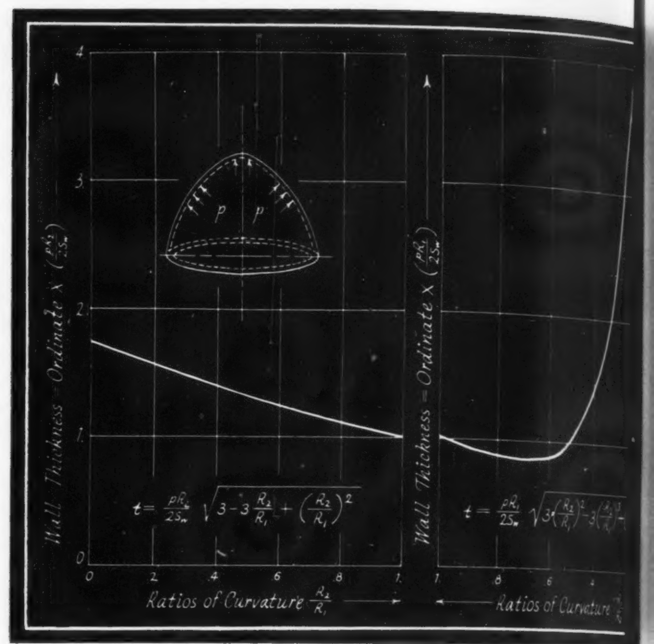


Fig. 3—Variations in wall thickness for cylinder head of general surface of revolution

termined because, in a hemispherical shell, the two stresses S_1 and S_2 are equal.

Example 4—Conical Head: For a conical head, as shown in *Fig.* 4, Equation *c* can be used for the stresses by noting that $R_1 = \infty$ and $R_2 = h \tan \alpha$. That is,

$$S_1 = \frac{pR_2}{2t}, \qquad S_2 = \frac{pR_2}{t} \qquad (e)$$

Using Equation 1 the distortion energy is

$$V = .75\left(\frac{pR_2}{t}\right)^2 c \qquad (f)$$

Equating this energy to the allowable value,

$$t = \frac{pR_2}{S_w}\sqrt{.75} \qquad (g)$$

The maximum value of t is for the element where R_2 is maximum. That is,

$$(R_2)_{max} = \frac{r}{\cos\alpha} = \frac{r}{h}\sqrt{h^2 + r^2}$$

Then the required wall thickness is

$$t = .87\frac{pr}{S_w}\sqrt{1 + \left(\frac{r}{h}\right)^2} \qquad (5)$$

Value of the wall thickness t is dependent upon the dimension ratio r/h. The variation in the thickness is shown in *Fig.* 4 for all possible values of r/h.

[2] For the stress analysis of this problem see, for example, S. Timoshenko—*Theory of Thin Plates and Shells*, McGraw-Hill Book Co.

Example 5—*Ellipsoidal Head:* For a pressure vessel or head of ellipsoidal shape, as shown in *Fig.* 5, the stresses have been shown to be the following[3]:

$$S_1=\frac{pka}{2t}v \quad \text{and} \quad S_2=S_1\left(2-\frac{1}{v^2}\right) \dots\dots\dots\dots (h)$$

where $k=a/b$ and $v=[1+(k^2-1)cos^2\beta]^{-1/2}$

Following the procedure as previously used, the distortion energy for the stressed element is

$$V=c\left(\frac{pka}{2t}\right)^2\left(3v^2-3+\frac{1}{v^2}\right) \dots\dots\dots\dots (i)$$

The critical element will be where V is maximum as v varies. By calculus this is defined by $dV/dv=0$ or $v=.76$. Placing this value of v in Equation *i* the maximum values of V is

$$V_{max}=.115c\left(\frac{pka}{t}\right)^2 \dots\dots\dots\dots (j)$$

Equating this energy to the allowable value $cS_w{}^2$, the wall thickness required is

$$t=.34\frac{pa}{S_w}k \dots\dots\dots\dots (6a)$$

The maximum value of V may also be for $\beta=0$ or $\beta=90°$. For these values the wall thickness value is

$$t=.50\frac{pa}{S_w}\sqrt{3-3k^2+k^2} \dots\dots\dots\dots (6b)$$

[3]See, for example, "The State of Stress in Full Heads of Pressure Vessels"—W. M. Coates, *Transactions,* A.S.M.E., Vol. 52, 1930, Paper APM—52—12.

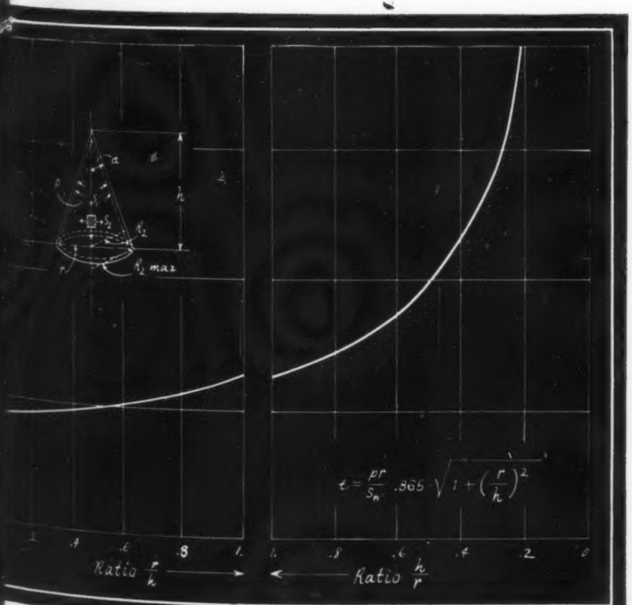


Fig. 4—Wall thickness for a conical cylinder head

or

$$t=.50\frac{pa}{S_w}k \dots\dots\dots\dots (6c)$$

The value given by Equation 6*c* is greater than that given by Equation 6*a*. A plot of Equations 6 is shown in *Fig.* 5. From this diagram the governing value of the wall thickness is then the maximum ordinate to the curves shown.

Example 6—*Torus:* A special type of pressure vessel is the torus[2] in *Fig.* 6. In the wall of this vessel, at any point at a distance r_0, the stresses are

$$S_1=A+\frac{Ab}{r_o} \quad \text{and} \quad S_2=A \dots\dots\dots\dots (k)$$

where $A=pa/2t$.

The distortion energy for this element by Equation 1 is

$$V=cA^2\left(1+\frac{b}{r_o}+\frac{b^2}{r_o{}^2}\right) \dots\dots\dots\dots (l)$$

Placing $dV/dr_0=0$, the critical element is at $r_0=b-a$. Maximum value of the energy V is

$$V_{max}=cA^2\left[1+\frac{b}{(b-a)}-\frac{b^2}{(b-a)^2}\right] \dots\dots\dots\dots (m)$$

Equating this energy to the allowable value $cS_w{}^2$, the wall thickness required is

$$t=\frac{pa}{2S_w}\sqrt{1-\frac{1}{1-r_d}+\frac{1}{(1-r_d)^2}} \dots\dots\dots\dots (7)$$

where $r_d=a/b$. *Fig.* 6 shows the variation in the thickness t for different values of r_d.

TANKS: Tanks of various shapes are used for storage of liquids. The tanks considered are open at the top and the liquid is subjected only to the pressure of its weight. There are a number of considerations in the design of these tanks—such as the design of the riveted or welded joints and of the supporting structure. Another is the effect of the membrane stresses due to the weight of the liquid. For example, in the conical tank in *Fig.* 7, the membrane stresses are

$$S_1=\frac{y\tan\alpha\left(d-\frac{2}{3}y\right)\gamma}{2t\cos\alpha} \dots\dots\dots\dots (n)$$

$$S_2=\frac{\gamma(d-y)y\tan\alpha}{t\cos\alpha} \dots\dots\dots\dots (o)$$

Using Equation 1, the distortion energy for the element at A is

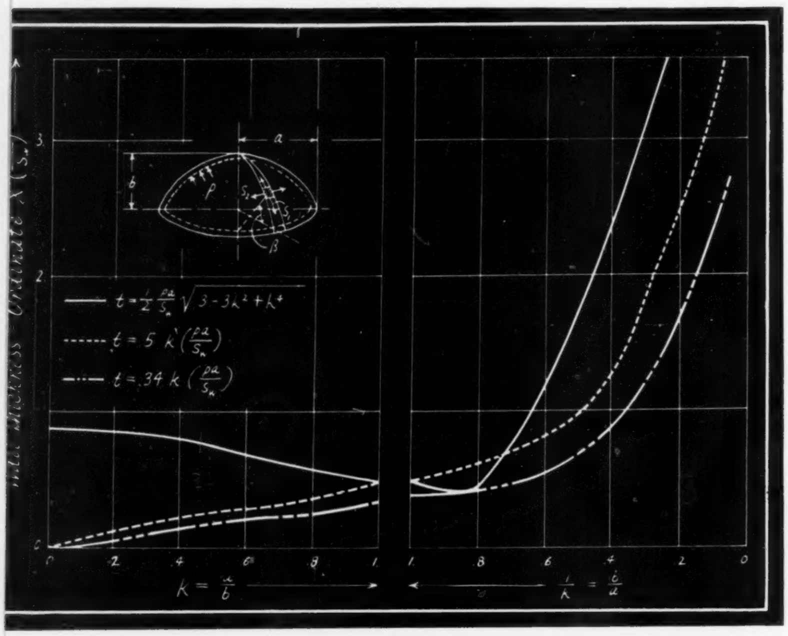


Fig. 5—Wall thickness for a head of ellipsoidal shape

$$V=c\left[\frac{y^2}{4}\left(d-\frac{2}{3}y\right)^2-\frac{y^2}{2}\left(d-\frac{2}{3}y\right)(d-y)+y^2(d-y)^2\right]\frac{\gamma \tan \alpha}{t \cos \alpha}$$

The maximum value of V is for $dV/dy=0$, or for

$$y=.52d$$

For this value of y

$$V_{max}=c\frac{\gamma \tan \alpha}{t \cos \alpha}.05d^4 \quad \ldots\ldots\ldots\ldots\ldots\ldots \text{(p)}$$

Placing this value of the distortion energy equal to the allowable value cS_w^2, the wall thickness required becomes

$$t=\frac{\gamma \tan \alpha}{S_w \cos \alpha}.225d^2$$

or

$$t=\frac{\gamma d^2}{S_w^2}.225\frac{a}{d}\sqrt{\left(\frac{a}{d}\right)^2+1} \quad \ldots\ldots\ldots\ldots\ldots\ldots (8)$$

Fig. 7 shows the variation in the wall thickness for different ratios of a/d and d/a from 0 to 1. This gives values for all possible angles of α.

PIPING: New processes in the power, oil refining and chemical industries have lead to larger units and higher operating pressures and temperatures. This increase in size has made it increasingly important to analyze the forces and stresses produced in piping systems. The stresses are produced by expansion due to temperature changes, internal pressure and dead weight. Expansion joints of different types sometimes are used to provide for effects of thermal expansion produced by hot fluids. It is desirable, however, to eliminate unnecessary anchors and other restraints for an economical piping design. Determination of the forces acting on a general pipe system is a complicated one since the reactions produced are statically indeterminate and the structure must be considered as a three-dimensional space problem. Analysis of these statically indeterminate reactions has been studied by many engineers[4]. The problem of the *combined stress effect* produced in piping due to axial loads, bending moments, internal pressure and twisting moments, however, has not been adequately considered.

In *Fig.* 8 is represented a pipe subjected to the forces previously mentioned. The piping discussed

Fig. 6—Right—Thickness of wall for a pressure vessel in the shape of a torus

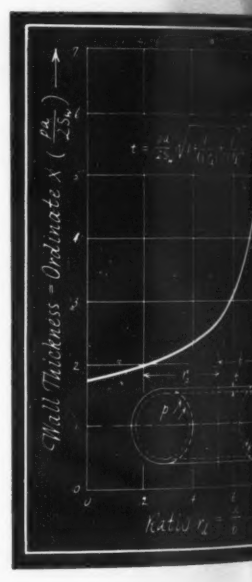
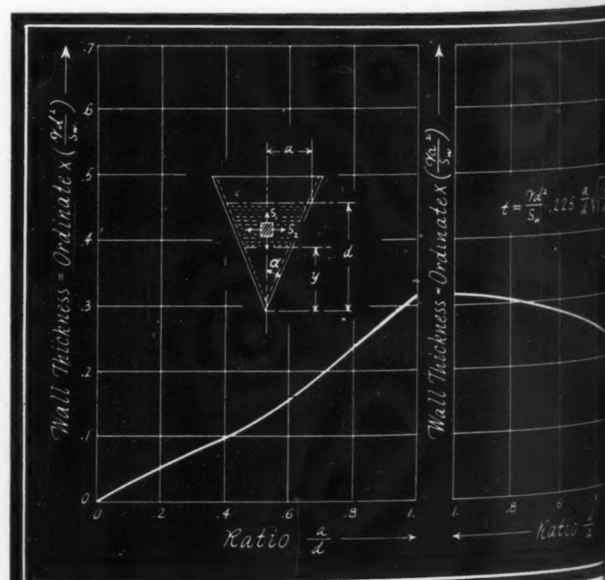


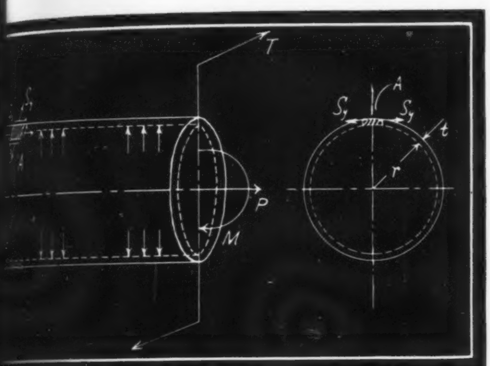
Fig. 7—Below—Wall thickness for conical tanks

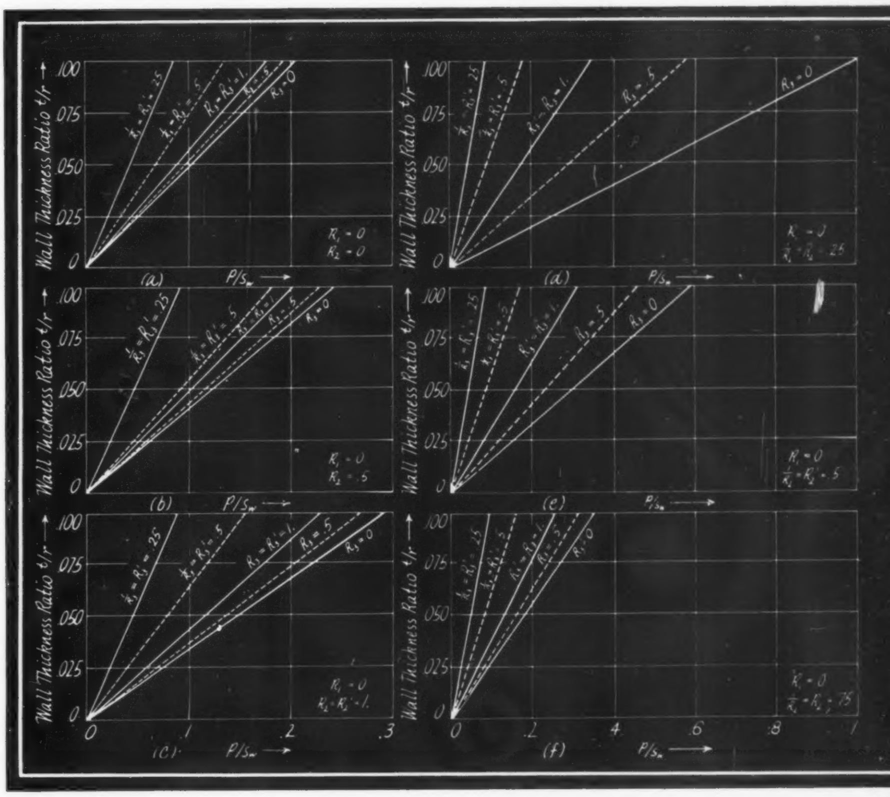


[4] For an excellent analysis of this problem see, *Design of Piping Systems—Expansion Stresses and Reactions in Piping Systems*, Published by the M. W. Kellogg Co., Jersey City, N. J., 1941. This manual also gives a good bibliography on these studies.

Fig. 8—Below—Stresses on an element in pipe are produced by bending moment, axial load, internal pressure and twisting moment

Fig. 9—Right—Pipe thickness for given values of R_3 and R_3' with $R_1=0$ and R_2 varying





for the present will be for relatively low pressures so that the wall thickness is small compared to the pipe diameter. In general, the effects of transverse shear forces can be neglected. Bending moments may act in two planes at right angles but it is only necessary to consider the resultant bending moment M.

Stresses produced on an element at A in *Fig.* 8 are a normal stress S_x from the bending moment and axial load P, a normal stress S_y due to the internal pressure p, and a shear stress S_s produced by the twisting moment T. The values of these stress components for the critical element at the outer top fiber are

$$S_x=\frac{M_r}{I}+\frac{P}{A};\qquad S_y=\frac{pr}{2t};\qquad S_s=\frac{Tr}{J}\qquad\ldots\ldots(q)$$

In these equations P represents an axial load which may include the longitudinal load produced by the internal pressure. For a thin-walled tube the values of the moment of inertia I, polar moment of inertia J and area A can be approximated to the following values:

$$I=\frac{\pi}{4}[(r+t)^4-r^4]\approx\pi r^3t\qquad\ldots\ldots(r)$$

$$J=\frac{\pi}{2}[(r+t)^4-r^4]\approx2\pi r^3t\qquad\ldots\ldots(s)$$

$$A=\pi[(r+t)^2-r^2]\approx2\pi rt\qquad\ldots\ldots(t)$$

Placing these values of I, J and A in Equations q, the stress components are

$$S_x=\frac{M}{\pi r^2t}+\frac{P}{2\pi rt};\quad S_y=\frac{pr}{t};\quad S_s=\frac{T}{2\pi r^2t}\qquad\ldots\ldots(u)$$

The expression for the distortion energy represented by the above stresses can be obtained by using Equation 1 and replacing S_1 and S_2 by their values in terms of S_x, S_y and S_s; namely,

$$\left.\begin{matrix}S_1\\S_2\end{matrix}\right\}=\frac{S_x+S_y}{2}\pm\sqrt{\left(\frac{S_x-S_y}{2}\right)^2+S_s^2}\qquad\ldots\ldots(v)$$

Making this substitution

$$V=c(S_x^2-S_xS_y+S_y^2+3S_s^2)\qquad\ldots\ldots(1a)$$

The value of V in terms of the loads is found from Equations u and $1a$. Its value is

$$V=c\left(\frac{1}{2\pi r^2t}\right)^2[(2M+Pr)^2-(2M+Pr)(\pi pr^3)+(\pi pr^3)^2+3T^2]\qquad\ldots\ldots(w)$$

Equating this energy to the allowable value cS_w^2 and solving for the wall thickness ratio t/r

$$\frac{t}{r}=\left(\frac{p}{S_w}\right)\sqrt{.10(R_1+.5R_2)^2-.16(R_1+R_2)+.076R_3^2+.25}\qquad\ldots\ldots(9)$$

where R_1 R_2 and R_3 are

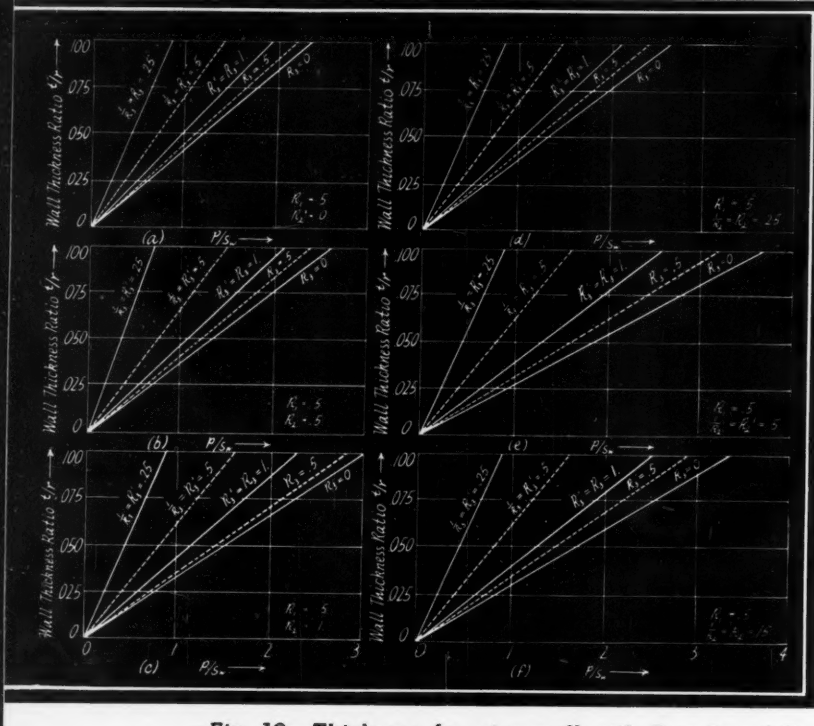
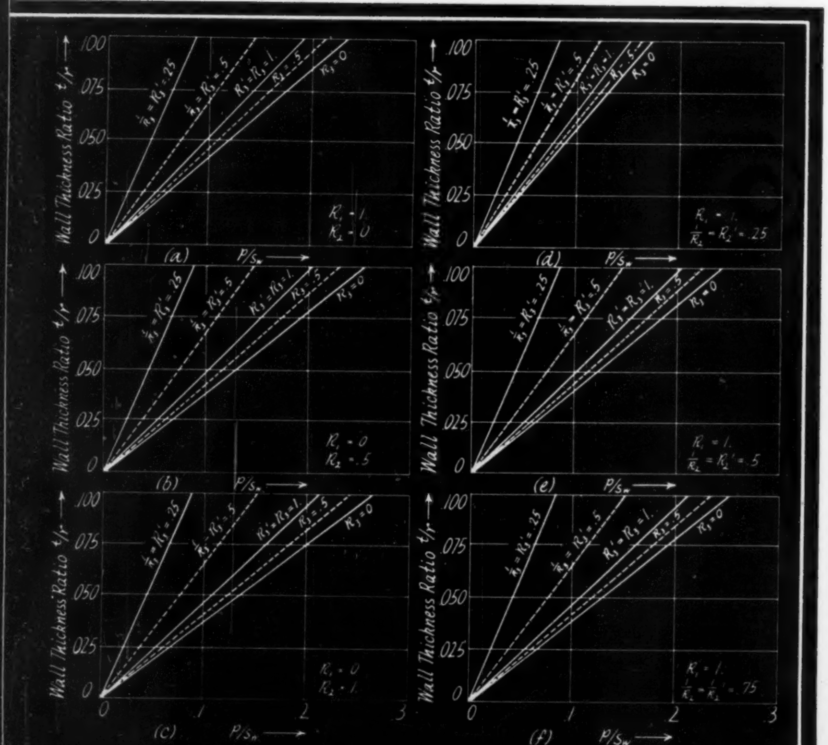


Fig. 10—Thickness for pipe wall with $R_1=.5$ and R_2 varying as shown in the lower right-hand corner of each chart

Fig. 11—Pipe thickness for values of R_1 and R_2 as shown in the lower corner of each chart



$$R_1=\frac{M}{r^3p}; \qquad R_2=\frac{P}{pr^2}; \qquad R_3=\frac{T}{pr^3} \quad \ldots\ldots\ldots\ldots(x)$$

That is, the pipe thickness t is given in Equation 9 in terms of the working stress S_w and the various load ratios R defined in Equations x. Design charts representing the variation in the wall thickness for different combinations of the forces acting are given in *Figs.* 9 to 11. An inspection of these diagrams shows that the wall thickness required is sometimes appreciably different from the values obtained when using the usual stress theory.

It should be emphasized that only one of many factors influencing the design of piping has been considered. Influence of creep and the determination of deflections and rotations are sometimes important considerations. The value of the working stress S_w must also be selected in many cases with an allowance for the effect of temperature.

Conserves Critical Materials

To conserve raw materials and fully utilize scrap, raw materials and parts, leading manufacturing interests of Erie, Pa., have developed a plan of industrial conservation, is producing excellent results.

An executive committee formulates instructions for men in each plant heading up salvage departments. Also, it furnishes technical advice to indicate proper collection, segregation and disposal of scrap by classification into iron, steel, nonferrous, etc. Data pertaining to all government regulations as well as listings of buyers, dealers, brokers or consumers of scrap according to established classifications are made available to each manufacturer.

Other useful information data will be distributed, and records of scrap collected and salvaged by each plant assembled and evaluated. For future guidance, studies will be made on the practicability of extending practices and procedures which have proved profitable in certain organizations to others operating on comparable bases. Also, the feasibility of adapting successful methods of large plants to smaller ones will be considered. Reasons for undue accumulation or sluggish movement of waste materials will be analyzed in an effort to increase the effective use of strategic materials.

Possibilities of transferring equipment not needed in one plant for production in another will be carefully weighed. For consultation purposes, services of an experienced salvage engineer are available on specific problems. Speakers for important meetings and clearing house functions for information and plans for procedure are additional features of the program.

Training of salvage men in definition of scrap, its nature, handling and salvage is included in the program. In each plant, these men arrange their own meetings and are responsible for conservation.

Fig. 1 — Right — Pneumatically operated typewriter for duplicating letters from rolls. Containing two rolls, the machine may be preset by pushbuttons to type any desired number of combinations

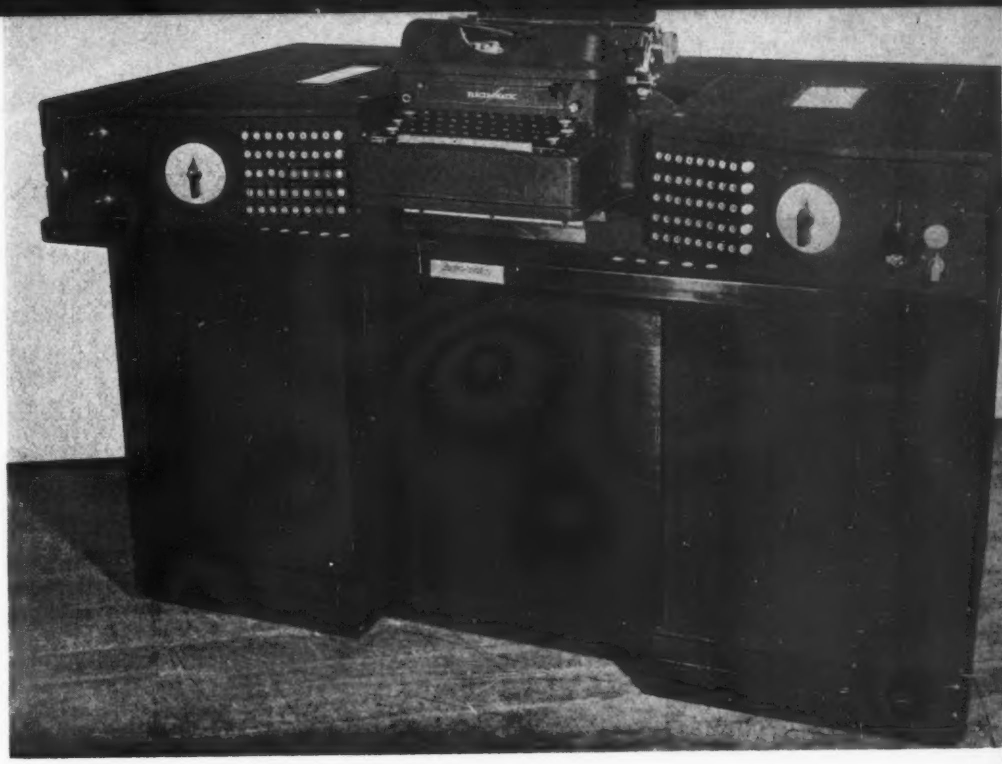
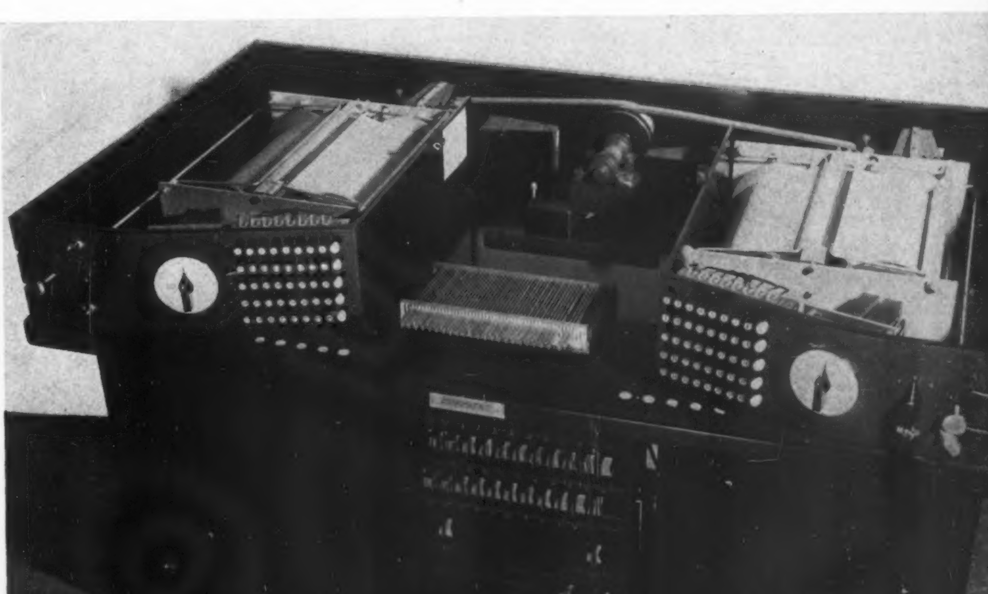


Fig. 2—Below—Main action unit showing links connected from "pneumatics" to key-actuating lever unit and two records in position over tracker bars of machine

By H. W. Kloid
Development Engineer
American Typewriter Co.

Dual Pneumatic Drive Speeds Automatic Machine



AUTOMATIC operation of a typewriter to utilize pneumatic impulses governed by perforations on a record similar to a player-piano roll provides quiet and dependable operation, long life, and flexibility that makes the machine shown in *Fig.* 1 suited to applications where a portion of the material being typed is repetitive. A means of selecting from several form letters or paragraphs on two given record rolls provides a method of preselecting a portion from one roll while the machine is actuating the typewriter from the other.

Preselection by means of 40 electric pushbutton switches for each record roll, *Fig.* 2, offers advantages of faster and more accurate operation than is possible with only one record roll

on machine. While normally referred to as an automatic typewriter, it is actually an accessory, and may be used in conjunction with any conventional or motor-driven typewriter.

Principal actuating mechanism is a "pneumatic," consisting of one stationary and one moving part hinged together and held apart by a cone-shaped coil spring, and covered by rubber cloth, *Fig.* 3. Power is produced by establishing a sub-atmospheric condition in the pneumatic through a suction pump. The power produced by the pneumatic is related to the surface of its top and the amount of suction used.

Connecting link between the pump and pneumatic is a pneumatic valve. Its function is to connect the pneumatic either with the outside air, establishing a pressure within the pneumatic equal to the atmosphere, or to link the pneumatic with the suction supply and collapse its movable part under the influence of sub-atmospheric pressure on the inside. The actuating means of the valve is a flexible diaphragm which rises under the influence of outside air pressure on the bottom, and suction above. Control means is an air impulse received through a perforation in the record, connected to the valve by a tracker bar hole and a rubber tube.

A bleed or vent is provided for connecting the suction channel above the diaphragm with the compartment below. This makes possible return of the diaphragm to its collapsed position when outside air is cut off, returning the valve to normal position. This open the suction channel to outside air through the vent. Bleed screws for pneumatics are placed to be readily accessible for adjustment, or to clean out foreign material that might interfere with normal operation.

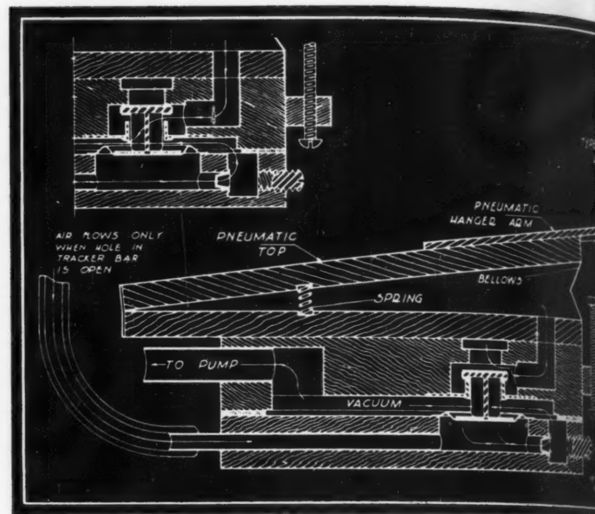
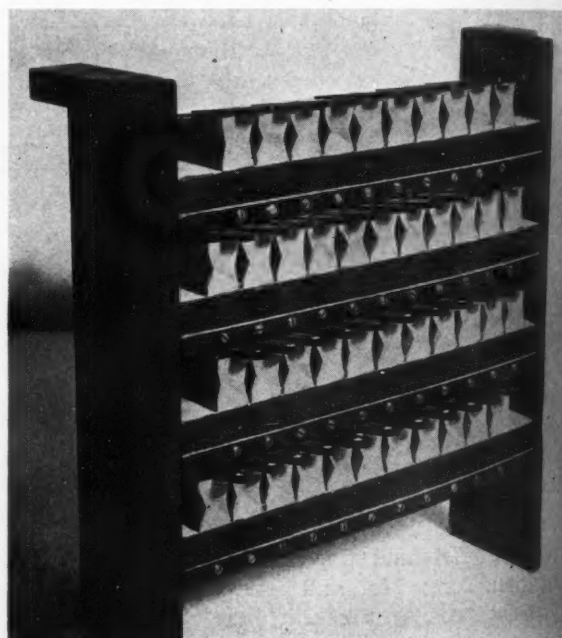


Fig. 3—Above—Cross section of main action bank pneumatic showing valve and tracker bar in closed position. In insert valve and tracker bar are open

Fig. 4—Below—Main action unit showing four banks of pneumatics and valve bleed screws. Collapse of a pneumatic operates the corresponding typewriter key



Selector Device Provides Versatility

The typewriter is actuated by four rows of pneumatics, *Fig.* 4, arranged corresponding to a typewriter key board, with shift, tabular key and back space pneumatics mounted under those for the letters. This unit is mounted directly below the typewriter position, while a larger pneumatic located to the right of the main action unit provides a means of line spacing and return of the typewriter carriage. Wires provide direct connections between typewriter key bars and the corresponding pneumatics. In operation, collapse of a pneumatic pulls down the key from below the typewriter, thus causing typing of corresponding character.

The original Auto-typist was limited in application to typing a single form letter, necessitating a record change every time a new letter was desired. Development of a selector device made it possible to choose several letters from a single record, or to compose a letter from a large number of available paragraphs. The need for even greater record capacity with the same selectivity brought development of a dual model, doubling the record capacity by employing two record spool mechanisms.

Two record rolls make necessary the incorporation of slide valves to prevent actuation of the typewriter from more than one record at a time. The slideable or upper plate of the slide valve, or tracker bar shut-off block, *Fig.* 5, has a series of nipples connected to the corresponding nipples on the tracker bar. The lower or stationary part of the slide valve is provided with the same number of nipple outlets, which connect by means of rubber tubing with a corresponding nipple on the other tracker bar shut-off block. This tube connection is interrupted by a T-nipple, the open end

of which is connected to the corresponding pneumatic. By a suitable perforation in the record, each block is shifted automatically in its off or closed position before the typewriter starts typing.

Opening of the slide valve is obtained by pressing a "start writing" button, which also makes the stop impulse of the record ineffective, thereby starting movement of the record. The other record cannot be started while the slide valve from

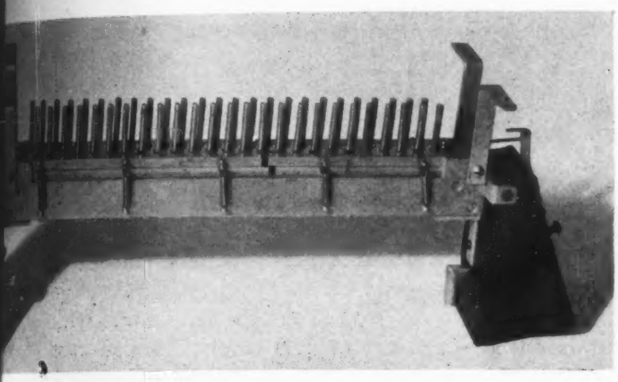


Fig. 5—Above—Tracker bar shut-off valve block showing actuating pneumatics at each end

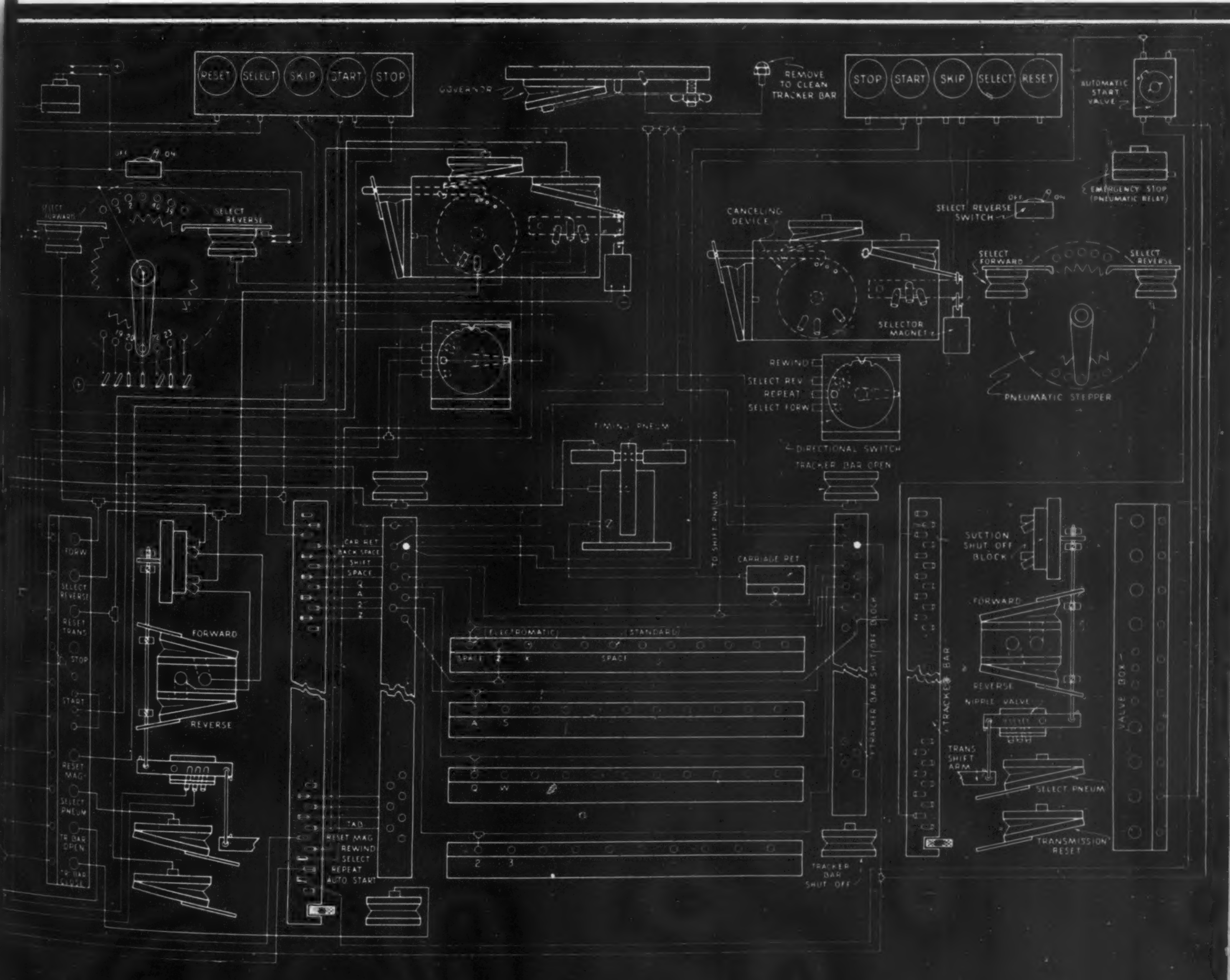
Fig. 6—Below—Schematic of pneumatic system for machine shows connections to all operating parts

the first tracker bar is passing impulses from its tracker bar. An automatic blocking device is employed to keep this slide valve in the closed or off position as long as the opposite slide valve is open. On completion of the paragraph or letter, the opened slide valve returns to the closed position, releasing the blocking device so that either tracker bar shut-off block may be opened when a start writing button is depressed for a new selection. A schematic of the pneumatic system is shown in *Fig.* 6.

The transmission clutch for each unit has three positions. In one position, the transmission moves the record forward in high speed to gain desired selection in minimum time. In another position, the transmission reverses at high speed to permit repeat of a form letter, or to speedily reach a selection punched earlier on the record. At an intermediate position, the record moves forward at desired typing speed.

Transmission positions are obtained by gear arrangements, with the shift lever actuated by pneumatics. These pneumatics are controlled from the tracker bar, or manually by the operator. Transmissions are driven by V-belts connected to an electric motor.

Shown in *Fig.* 7 is a square suction pump, con-



sisting of four pneumatics with intake valves connected to a common channel within a square frame. This supplies the vacuum necessary to actuate all pneumatic functions of the machine. The pump is rated at 15 centimeters of mercury suction, which during normal operation is vented or "leaked" to operate between 9 and 11 centimeters. Leakage is eliminated, however, when the carriage return operates to provide maximum suction

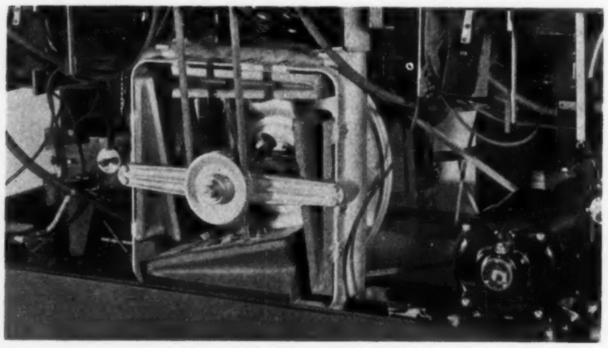
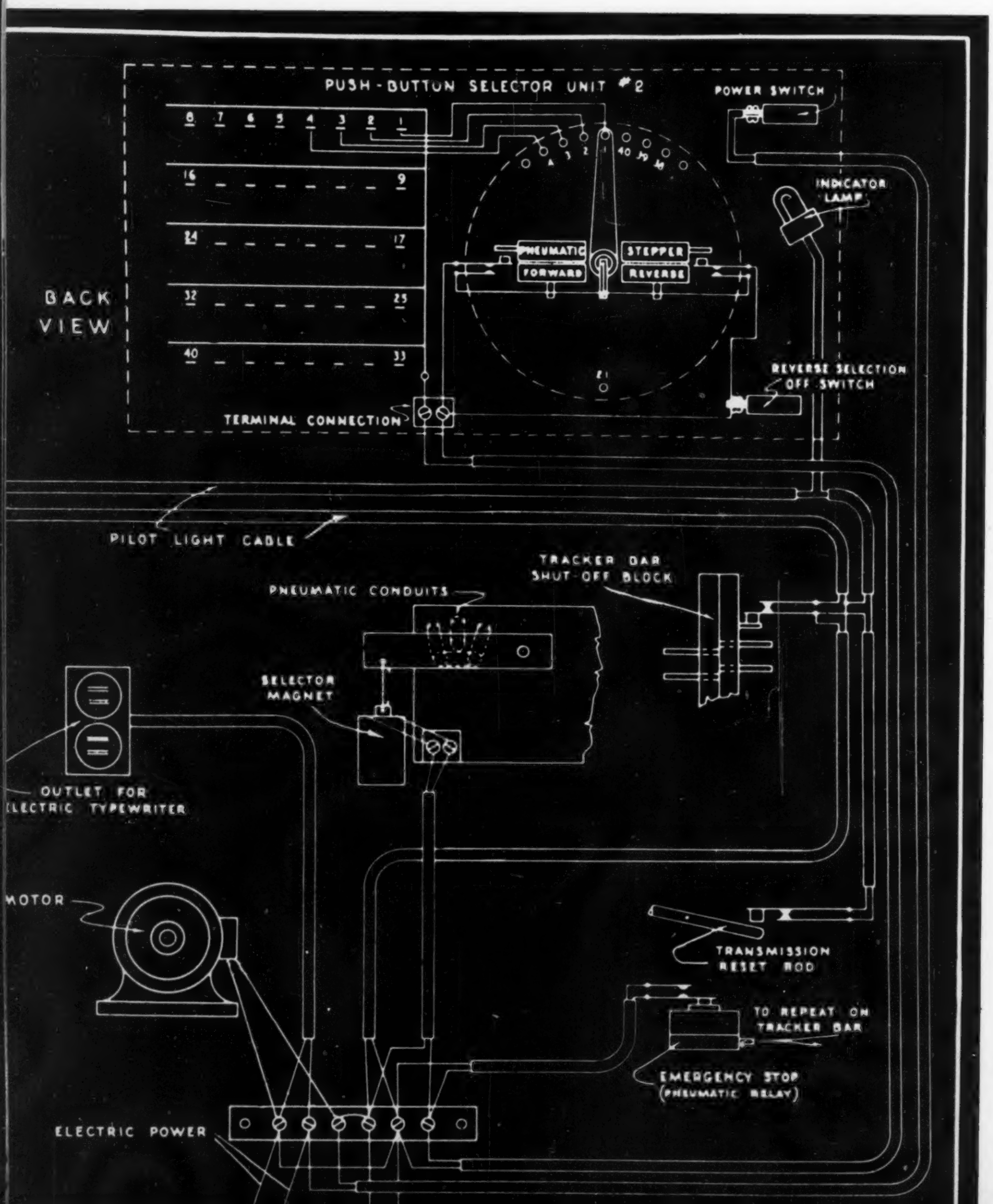


Fig. 7—Above—Rear assembly with pump and motor drive in foreground

Fig. 8—Below—Schematic diagram of right half of electrical control system. Other half is duplicate



from the pump for speedier return of the carriage.

Pump is driven by a ¼-horsepower split-phase, 1140 rpm. motor, either alternating or direct current, resilient mounted to eliminate hum or vibration. Other possible methods have been utilized to eliminate vibration and noise that would be objectionable in a business office. The result is that the only perceptible noise is the click of the typewriter keys themselves.

Pneumatics in the main action unit, connected to the typewriter keys, require power equivalent to three ounces of pressure to pull the key of an electric typewriter, and 12 ounces for a conventional typewriter. Regulation of the transmission for normal forward typing speed is accomplished by a rubber friction wheel movable on an intermediate shaft in the pump. This wheel presses against a rubber disk to gain speed of record in accordance with speed of the typewriter. Average typing speed is 110 to 125 words per minute with conventional typewriters and about 15 per cent higher when the machine actuates an electric motor driven typewriter.

Two pushbutton selector devices are incorporated in this machine, one for each record. Each device consists of a contact stepper unit, a record directional switch, a reverse selection cut-off switch, an emergency stop unit or pneumatic relay, a reverse selection canceling and selector magnet unit, 40 electric pushbuttons with reset buttons, and a suction cut-off block for transmission forward and reverse select pneumatics. The wiring diagram in *Fig.* 8 schematically shows the connections for these control elements.

The contact stepper unit is designed to move or step the electrical movable contact, forward or reverse, to a stationary contact that has an electrical connection with a corresponding pushbutton. The stepping of this movable contact is accomplished by two pneumatics, one forward and one reverse. The forward pneumatic has a direct connection with a select forward valve in the valve box, and steps once every time a series of select perforations pass over the tracker bar when the record is progressing in a forward direction. These select perforations provide subdivisions in the record roll, separating paragraphs, letters, or single words or phrases.

The reverse stepper pneumatic unit is interconnected through

(Continued on Page 134)

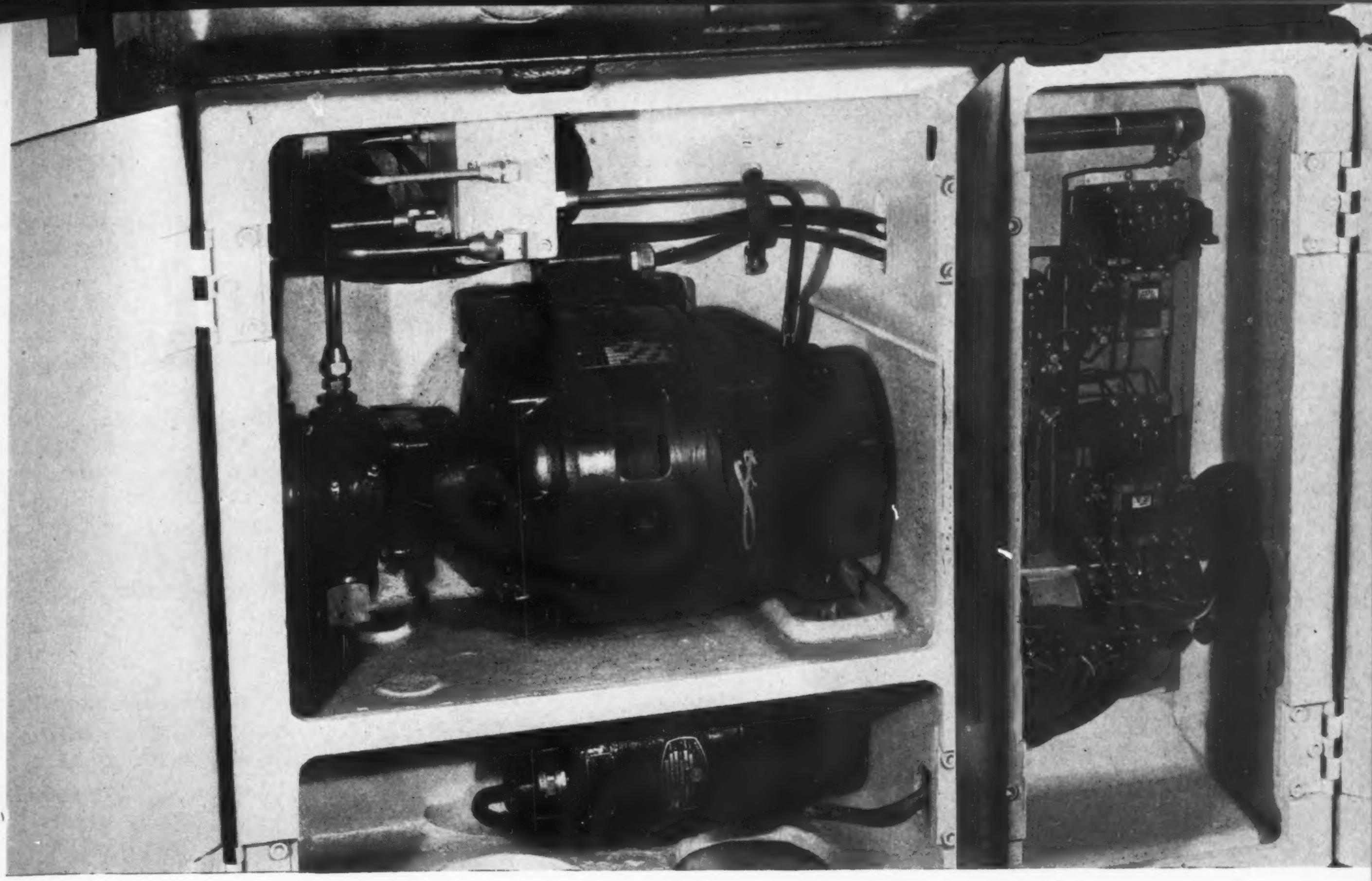


Fig. 1—Above—Built-in drives and control on grinder showing motor for hydraulic pump, direct-current generator for variable voltage headstock, and magnetic control. Wiring is prefabricated from cable and wiring channels are provided in bed casting

Build In Controls for Wartime Economy!

By R. S. Elberty

Electrical Engineer
Landis Tool Co.

BUILT-IN electrical drives and controls for production machines present many recognized advantages to the buyers and operators of these machines. Not so apparent are the advantages to the machine builder. A brief summary of these advantages might be tabulated as follows:

Advantages to the User

1. Self-contained machine, can be shifted in production line
2. No projecting controllers or conduits to be damaged by trucks
3. Electrical equipment is better protected from coolant, oil and chips
4. Floor space is reduced
5. Appearance is improved
6. Machine is easier to clean
7. From standpoint of electric shock, the machine is safer.

Advantages to the Manufacturer

1. Complete responsibility for design
2. Machine can be tested before shipment
3. Savings in cost of conduits and fittings
4. Reduced wiring and mounting time
5. A neater, more salable product
6. Painting is easier.

In order to build-in controls to gain these many advantages, the safety of the operator and the machine must be considered. For this reason the National Machine Tool Builders association has recently prepared and adopted electrical standards that might well serve as a reference to any machinery builder seeking a guide for good minimum

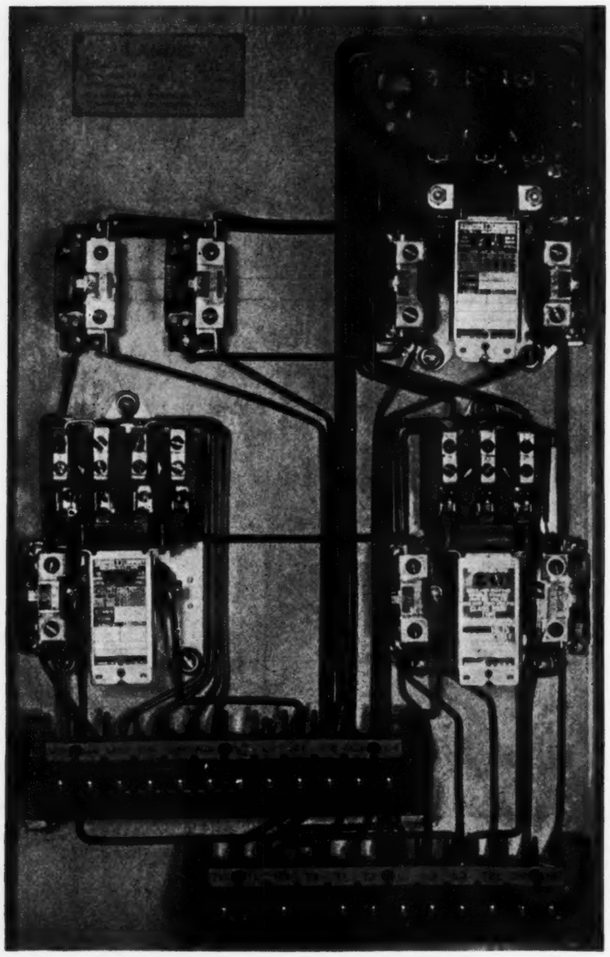


Fig. 2—Assembled magnetic control panel wired to reduce length of wire. Wires are color coded

practice in the application of built-in drives. Good design of built-in electrical control does not necessarily add to the cost of a machine. There are inherent economies in this type of application which the machine designer can utilize.

For example, built-in control is usually a special assembly of standard parts. A suitable starting point is in the control itself. This should be designed to use a minimum number of parts. The control criterion* may be applied to any control to determine if there are any parts in excess of those actually required. If so, the control may be redesigned to eliminate these parts. The designer should also determine if mechanically operated contacts can be substituted for magnetic relays in the control circuit. Limit switches, selector switches, etc., can often be substituted for magnetic contacts to good advantage.

Built-in controls usually permit the designer to use greater flexibility in the location of parts, as shown in *Fig.* 1. The first consideration should be maintenance of the machine, but next is the location of the control close to other electrical parts. This practice saves wire and reduces the time of installation, besides resulting in a good electrical job. The advantage of flexibility in the location of the equipment should not be abused. Its compartment should be suitable from every consideration, and not just any cored opening that happens to be in the machine.

Controllers may be divided into several parts and separately mounted to save space in the machine. This might be a convenient solution to a difficult design problem, but wiring time and material can be saved if the magnetic control is in one piece at one location in the machine. The same procedure in locating pushbuttons, limit switches, etc., will show economies in wiring time and savings in necessary war materials. Everybody gains from a well designed built-in control. The machine user has a simplified control, the machine builder reduces the cost of installation, and our nation saves critical materials.

Magnetic control panels, *Fig.* 2, offer economies in manufacture. For convenience in servicing and maintenance, front-of-board wiring is preferred.

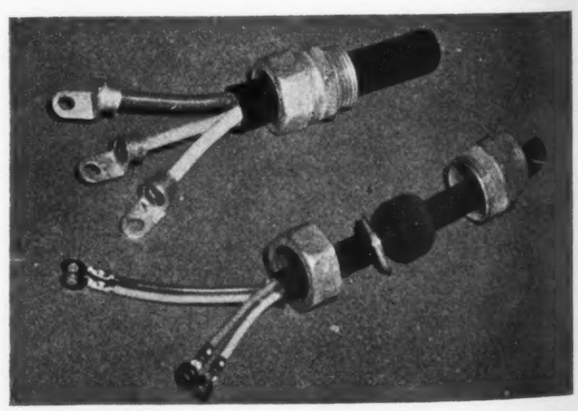


Fig. 3—Oiltight bushings effectively seal cables and clamp them to the machine part. Cables are color coded with terminals mechanically staked

*"Designing Control Circuits," MACHINE DESIGN, March, April and May, 1941.

MULTICONDUCTOR CODED CABLE*

Number of Conductors	Cable O.D. (limits in inches) A.W.G. Wire Size 14	12	10	8	6	4	Color of Cable Jacket	Color of Each Conductor: Red	Light Blue	Yellow	Orange	Purple	Navy Blue	Pink	Brown
2	.400-.410	....	....	....	....	....	Brown	X	X						
3	.420-.430	.465-.475	.515-.535	.680-.700	.830-.850	.935-.965	Black	X	X	X					
4	.455-.465	.490-.510	.565-.585	.745-.765	.885-.920	1.02-1.05	Navy Blue	X	X	X	X				
5	.500-.520	....	....	....	....	....	Purple	X	X	X	X	X			
6	.535-.555	....	....	....	....	....	Dark Gray	X	X	X	X	X	X		
8	.580-.600	....	....	....	....	....	Light Blue	X	X	X	X	X	X	X	X

*Each conductor has 19 strands covered with *SN* synthetic insulation. Cable is taped and covered with 3/64-inch jacket for Nos. 14, 12 and 10 size conductors. 8, 6 and 4 are covered with 4/64-inch jacket. Insulation thickness on each conductor is 2/64-inch for 14, 12 and 10, 3/64-inch for No. 8 and 4/64-inch for No. 6 and 4 conductor sizes.

In the machine tool industry, type *SN* wire† is coming into general use because of its resistance to the action of oil and coolant. This type of insulation has been greatly improved in recent years and it will probably come into more general use due to the scarcity of rubber and synthetic rubber. The material is very tough in resisting mechanical abrasion.

Many panels are also wired with *SN* wire, and the use of color wire to identify circuits has simplified servicing of complex control panels. The wires should not be placed on top of each other and crossovers should be avoided as much as possible. Wire is often wasted in the interest of neatness, and a return to a functional scheme of wiring will reduce the number and length of wires on most controllers. In these times there is no justification for the addition of materials for the sake of appearance only.

Many machine builders find they can assemble

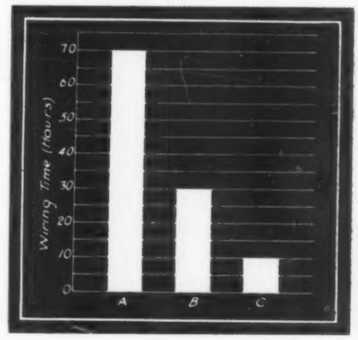
Fig. 4—Set of wires being prefabricated from type SN cable. In wiring the machine the electrician follows a color code and need not make soldered connections

their own special controllers at costs to compare favorably with market quotations on this type of equipment. Standard contactors and relays are built in large quantities and stocked by the electrical manufacturers. The machine designer has considerable freedom of choice in selecting standard items for assembly into special controls.

Various methods of applying wiring have been devised. The house-wiring system of pulling loose wires through conduits is being supplemented by methods that are more suitable to the machine and to the conditions under which the electrical equipment must function. Identification of wires by coding is being used increasingly, the old method of "ringing out" circuits being far from satisfactory on complex controls. Wires may be marked or numbered on each end, wiring harness can be prepared which cannot be incorrectly connected, or color coded wiring may be used. Any of these systems facilitates maintenance and simplifies the problem of machine wiring.

Cable connections to moving machine parts have always been a problem to machinery builders. Flexible metallic conduit will fail mechanically under severe flexing and it offers little or no protection from the action of coolant and oil. Cable with *SN* outer covering offers a practical solution to the problem and can be made oiltight and watertight through suitable fittings. While this is a new development, there are signs that this type of cable will become increasingly popular

Fig. 5—Time required to wire a grinding machine by three methods: (A) control on outside of machine with wiring in conduit, (B) built-in control using cabled wiring and (C) built-in control from prefabricated color-coded wiring



for different applications where wiring may be exposed to water or oil. It should not be used in high temperatures and it loses considerable flexibility when operated below the freezing point.

In any system of color coding, consideration must be given to the wire and cable manufacturing problem. Special wires and cables will have excessive costs unless purchased in large quantities. The accompanying table of color coded cable has been prepared with this thought in mind. This system has been in use by the Landis Tool Co. for over a year and the results have been quite satisfactory. Method of applying bushings for sealing and clamping the cable is shown in *Fig.* 3.

Prefabricated wiring systems save considerable time and prevent waste of material. One system of prefabrication is illustrated in *Fig.* 4. Size and length of wire or cable can be long enough to cover variations in dimensions, or mounting differences, but need be no longer. If the wire is cut to length away from the machine the work can be done by relatively unskilled labor, reserving skilled electricians for actual wiring of machines on the assembly line. Resulting savings in time and material will pay for the engineering time to lay out and specify a good job. *Fig.* 5 compares wiring time for control mounted outside, wired in conduit; built-in control, cable wired; and built-in, prefabricated wiring. Although a special case, the chart shows how assembly time can be reduced.

With our entire war effort calling for more and more production, machine designers can contribute man hours and strategic defense materials to national defense through applying built-in electrical control. Here are large possibilities, and the engineering doesn't necessarily call for great change.

† Synthetic insulated conductors, National Electric Code specification. Type *SN* insulation is defined in the 1940 code as "a flame-retardant, moisture-resistant approved solid synthetic insulation (type *SN*), with a temperature limitation of 60 degrees Cent. (140 Fahr.) in sizes Nos. 14 to 4/0 . . . . no outer fibrous covering being required".

Playing Hob with I

By John D. Howell
Remington Arms Co.

IN DESIGNING gear trains it is often felt that any departure from the standard 14½-degree involute tooth requires a special gear, and that such special gears are necessarily expensive. With modern methods of gear cutting, however, such forms as the 20-degree involute and even certain special gears of nonstandard form are not only no more expensive but also have advantages over the old 14½-degree form. It is hoped that this article will assist those machine designers who have only infrequent occasion to design trains of gears and who, therefore, may not be acquainted with the full possibilities of special gearing.

If gears are to be selected for the job it is highly desirable to specify the 20-degree full depth involute form rather than the 14½-degree. In fact, the British and other European gear standards, though recognizing the necessity for the 14½-degree form for replacement work, recommend that the 20-degree form be used on all new design. One can only assume that the persistent use of the 14½-degree gear is attributable to the attitude, "The old way has worked for fifty years. Why change?"

To answer this question, examination of *Fig.* 1 shows the teeth of a rack, a 24-tooth gear, and a 12-tooth pinion. *Fig.* 1*a* illustrates the 14½-degree involute and *Fig.* 1*b*, the 20-degree involute forms. The 20-degree rack, with its wider base, is the stronger form. Also, in the case of the 24-tooth gear, not only is the 20-degree the stronger tooth but, in addition, in the 14½-degree form the fillet between the teeth has cut a small amount out of the base of the involute itself. In the 12-tooth pinion even the 20-degree tooth has begun to develop a slight amount of undercut. In contrast the undercut of the 14½-degree, in addition to seriously weakening the base of the tooth, extends almost up to the pitch line, removing an important portion of the working surface.

In the 14½-degree form undercut will be present when the number of teeth is less than 32, may be excessive when the number of teeth is less than 22, and with 14 or less teeth, correct gear action is impossible without modifying the tooth form. In the 20-degree form, undercut is not present until the number of teeth is less than 18 and is not likely to be excessive until it is less than 14. As an example may be cited a design in which two 12-tooth, 14½-degree pinions were meshed together, giving a contact ratio, due to undercut, of .034! Contact ratio being a measure of the amount of gear action, there was practically no correct gear action at all. If these pinions had been of the 20-degree tooth form the contact ratio would have been 1.049, still by no means as large as desirable, but at least a ratio which could be expected to give continuous gear action.

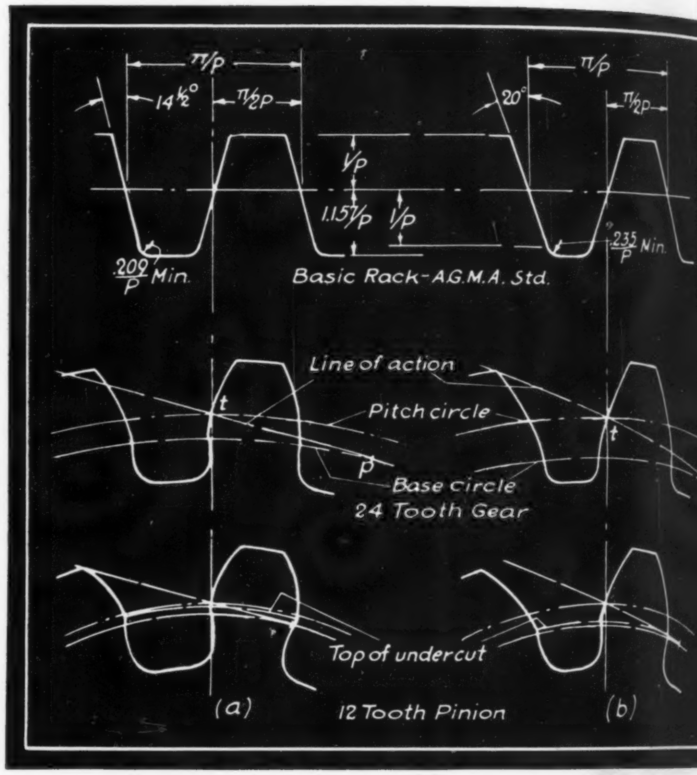


Fig. 1—Less undercut of teeth in the 20-degree pressure angle system (right) results in increased strength and contact ratio

Affects Contact Ratio

From this example it will be noted that, in addition to weakening the tooth, undercut may also have an adverse effect on the contact ratio. Thus, while the 14½-degree form can have a greater contact ratio in the larger gears than can the 20-degree (the limiting values, i.e., the contact ratio for a rack with an infinitely large gear, being 2.63 for the 14½-degree and 1.98 for the 20-

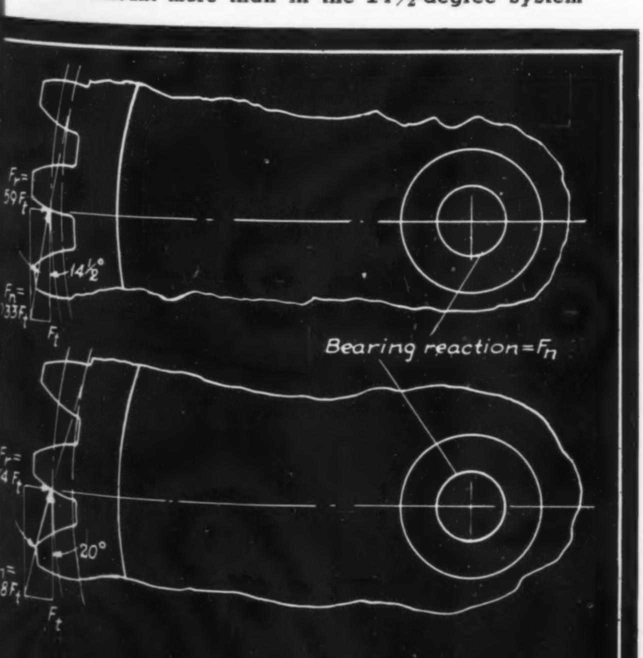
Ie Gears!

degree), the presence of undercut in the smaller and often more useful pinions is likely to cause a far smaller contact ratio in the 14½-degree system.

Another point in which the 20-degree form is superior to the 14½-degree is tooth strength. A tooth can fail in three ways: The first, already noted, is failure as a beam, in which the form is a controlling factor; the second is by spalling, that is by portions of the contact surface flaking off from excessive pressure; and the third is by wear. Resistance to the second is a function of the radii of curvature of the contacting surfaces and, as can be seen from *Fig.* 1, where the line *pt* is the radius of curvature, the 20-degree form has the larger value. In addition, this larger radius of curvature helps the third condition of failure by distributing the pressure on the lubricant film over a larger contact area, thus making it easier to maintain this film unbroken.

The only advantage that the 14½-degree form does have over the 20-degree is that there is less radial thrust or "separating force" between two gears. Far too much importance, however, is usually given to this advantage. *Fig.* 2 shows the force systems of the two types of gearing. While it is true that the radial force F_r is comparatively larger in the 20-degree system it will be noted that the load on the shaft and consequently on the bearings, which is the important one in design considerations, is the resultant, F_n, of this force and driving force F_t, and that this resultant is very little greater in the 20-degree system. To be exact the resultant force is greater than the driving

Fig. 2—Bearing reaction in the 20-degree system (lower) exceeds the driving force by only a negligible amount more than in the 14½-degree system



force by 3.3 per cent in the 14½-degree and 6.8 per cent in the 20-degree, a difference which is negligible.

So far nothing has been said of the stub involute. This form is the same as the 20-degree full depth involute except that, as the name implies, a shorter tooth is used. This results in a tooth somewhat stronger in beam strength but decreases the wearing surface and also tends to reduce the contact ratio so that the action may not be as smooth.

As yet only the various standard forms of gearing have been discussed. To aid in understanding the design of more special gears it may be well to review the fundamentals of involute gearing. A pair of gears is often thought of as analogous to two cylinders rolling against each other, their

Nomenclature

P = diametral pitch
p = circular pitch, π/P
p_b = base pitch
c = standard distance between mating gears, $(N_1+N_2)/2P$
c_r = running center distance
ϕ = pressure angle
ϕ_r = running pressure angle, $\cos\phi_r = (c/c_r)\cos\phi$
R, D = pitch radius, diameter
R_o, D_o = outside radius, diameter
R_u, D_u = undercut radius, diameter, cutting below which will produce undercut
x = amount of involute profile removed by undercut
R_b+x = radius to top of undercut
N = number of teeth in gear
b = backlash, inches
m_p = contact ratio
y = radius of top of hob tooth
j = increase in depth of cut to give backlash, inches
β = angular backlash
h = depth of cut

diameters being equal to the pitch diameters of the corresponding gears. For the involute tooth form, a more informative analogy may be developed by considering two pulleys, with diameters equal to the *base* diameters of the corresponding gears, connected by means of a crossed belt. *Fig.* 3 illustrates this concept. The two shafts have centers at O' and O. The heavy circles are two pulleys and the crossed belt appears as the two straight lines tangent to these circles. If these pulleys are to be replaced by gears which will give the same speed ratio it can be seen that, by similar triangles, the pitch circles of the gears will pass through the point p where the two sides of the belt cross. Disks of cardboard are attached to the two pulleys and a pencil to the belt at point aa' so that this pencil will draw on both disks. As one pulley drives the other through the belt, the pencil will draw curve a' on the left-hand disk, and another curve a on the right-hand disk.

As these curves are drawn on the disks by the wrapping or unwrapping action of the belt from the respective pulleys, they will be involute curves

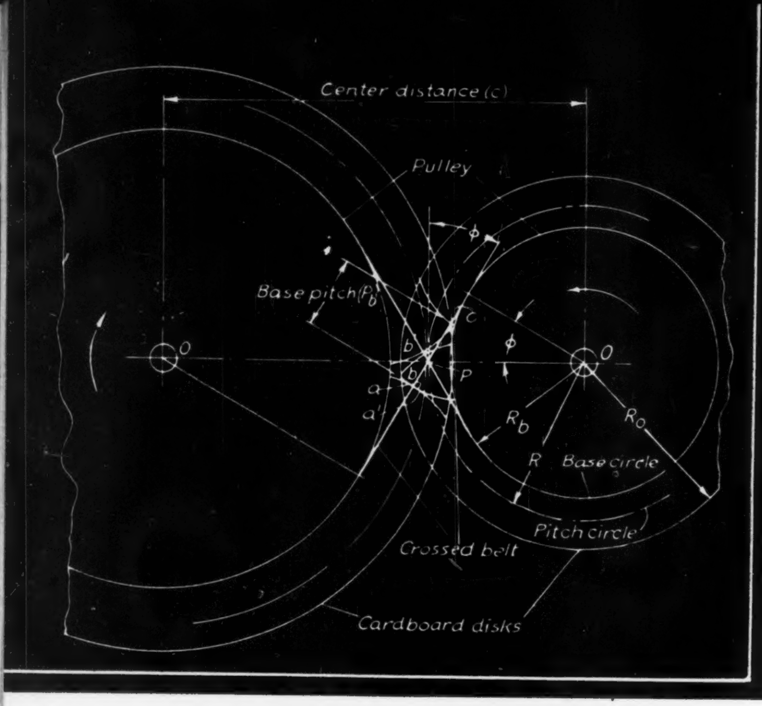


Fig. 3—Diagram of crossed-belt analogy illustrates the effect on gear properties of changing running center distances of mating gears

and, as can be seen, these curves would be satisfactory profiles for gear teeth. It can also be seen that when the two curves are in contact, the point of tangency will always be along the line of the belt. This line in a pair of gears is called the line of action and the angle it makes with a perpendicular to the line of centers is called the pressure angle, ϕ. If now the distance between o and o' is increased, lengthening the belt as necessary, the pencil will still draw the same involutes since the belt still has the same unwrapping action from the pulleys and the chief effect will be to increase the angle ϕ. It will also be noticed that the pitch radii Op and $O'p$ has been increased in the process, which shows an interesting fact about all involute gears—while the base circle is a fixed property of a given gear the pitch diameter and pressure angle are not. The relation between these three values[1], as can be seen from *Fig.* 3, is

$$D \cos \phi = D_b$$

[1] Subscripts 1 and 2 are used to distinguish between the same symbols applied to two gears of a pair.

[2] In the following formulas, the involute function of ϕ ($inv\ \phi$) may be found tabulated in Buckingham's *Manual of Gear Design*, Section I. It may also be calculated from the relationship, $inv\ \phi = \tan \phi - arc\ \ \phi$.

and for a pair of meshing gears

$$c \cos \phi = R_{b1} + R_{b2}$$

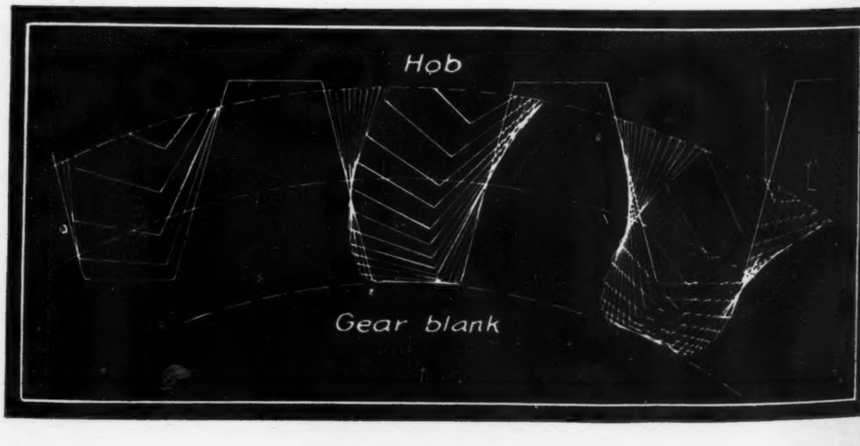
Another effect of this separation of centers can be seen if another pencil is attached to the other straight portion of the belt at point bb'. At the original centers this pencil will also draw involutes. If the centers are separated, the belt being suitably lengthened between the two pencils, the same curves will be drawn on the disks but they will be spaced apart by the amount the belt was lengthened. In gears then, increasing the center distances over the standard value will result in increased backlash, that is, in an increased distance between nondriving faces of the mating gear teeth. Or, to rephrase this definition, backlash is the thickness of a feeler gage that will just fit between nondriving faces on the mating gear teeth. The formula[2] for backlash produced by increasing the center distance between standard gears is:

$$b = 2c_r \cos \phi_r\ (inv\ \phi_r - inv\ \phi)$$
$$= 2(R_{b1} + R_{b2})\ (inv\ \phi_r - inv\ \phi)$$

If, as mentioned in the foregoing, the pitch circle is not a fixed property of a given gear, what becomes of the circular pitch, that is, the distance along the pitch circle between similar faces of successive teeth? This, too, must vary with the pitch circle. However, considering this next tooth face as drawn by another pencil fixed at c to the same portion of the belt as aa', it is evident that this pencil will remain a fixed distance from aa' as the belt travels from one pulley to another. This distance ac is called the base pitch and is a fixed property of the gear and of any gear that will mate with it.

Another term that has been used previously without definition is "contact ratio." Looking at the crossed belt diagram, *Fig.* 3, it will be noticed that there can be no contact between the curves except on that proportion of the line of action which lies on both disks. If the length of this portion were exactly equal to the base pitch one tooth would make contact with a tooth of the mating gear just as the preceding tooth lost contact, and thus would have a contact ratio of 1.

Fig. 4—Involute is generated by successive contact of a number of hob teeth in generating plane



Contact ratio, then, is the ratio of the length of the line of action to the base pitch. While correct gear action will be obtained with any contact ratio of 1 or over, it is generally considered desirable to maintain a ratio of better than 1.4 for smooth and quiet gear action. The formula[3] for contact ratio is

$$m_p = \frac{\textit{length of line of action}}{\textit{base pitch}}$$

$$= \frac{\sqrt{R_{o1}^2 - R_{b1}^2} + \sqrt{R_{o2}^2 - R_{b2}^2} - \sqrt{c_r^2 - (R_{b1}+R_{b2})^2}}{p_b}$$

$$= \frac{\sqrt{R_{o1}^2 - R_{b1}^2} + \sqrt{R_{o2}^2 - R_{b2}^2} - c \sin \phi}{p \cos \phi} \quad \ldots\ldots\ldots (49)$$

When undercut is present it may further reduce the contact ratio if the radius to the top of the undercut intersects the line of action inside the outside diameter of the mating gear. If this condition is present in only one gear, the contact ratio depends on this gear alone and is equal to

$$m_p = \frac{\sqrt{R_o^2 - R_b^2} - \sqrt{(R_b + x)^2 - R_b^2}}{p_b}$$

$$= \frac{\sqrt{R_o^2 - R_b^2} - \sqrt{2R_b x + x^2}}{p_b}$$

If the condition is present in both gears the contact ratio becomes

$$m_p = \frac{c \sin \phi - \sqrt{(R_{b1}+x_1)^2 - R_{b1}^2} - \sqrt{(R_{b2}+x_2)^2 - R_{b2}^2}}{p_b}$$

In any given case, the lowest of the three values is the correct contact ratio and should be used in determining this index to gear action.

If the number of teeth in a gear is allowed to increase indefinitely, the limiting form is a rack. The limiting form of the involute tooth profile at the same time becomes a straight line perpendicular to the line of action. As the line of action must always be perpendicular to the tooth profile the pressure angle of a rack is fixed and consequently the circular pitch is also fixed. Therefore, by a gear with a given pressure angle and circular or diametral pitch, is meant one which will run with a rack of this pressure angle and pitch. When the center of the gear is moved away from the rack, as indicated, the pressure angle cannot change and the pitch diameter remains the same, the pitch line of the rack moving outward with the gear.

Having a rack that will run with a given system of gears, it should be possible to use this rack to generate new gears of this system. Hobbing, the

[3] Number in parenthesis beside a formula indicates that the formula is taken from Buckingham's *Manual of Gear Design*, Section 2 and is so numbered therein. Formulas not otherwise credited have been derived by the author.

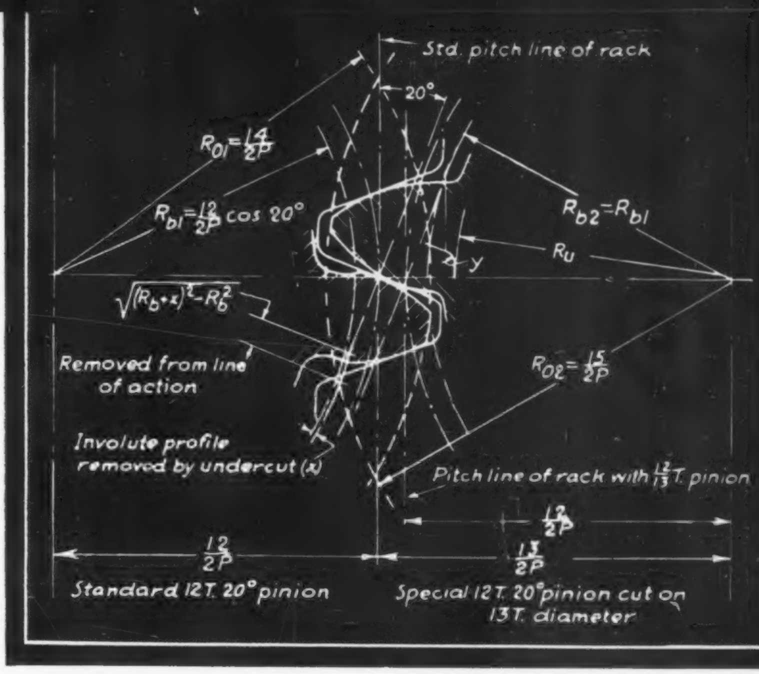


Fig. 5—Stronger tooth profile and larger proportion of working surface result from cutting 12 teeth on a 13-tooth gear blank

method of accomplishing this, is illustrated in *Fig.* 4.

Whereas the base pitch, tooth shape, pressure angle and circular pitch are fixed in the basic rack, as has been seen by the crossed belt analogy, they are not fixed in the gear generated from it. *Fig.* 5 shows, on the left, the tooth of a standard 20-degree, twelve-tooth pinion, and on the right, the tooth of a special twelve-tooth pinion cut with a standard 20-degree hob, but having the outside diameter of a thirteen-tooth pinion. Between the two gear teeth can be seen the form of the standard 20-degree rack. It may be assumed that this rack is the outline of the hob generating the left-hand pinion. Therefore, at the most, only three-quarters of the tooth face is useful for gear action, the remainder being below the base circle of the involute and therefore useless. In addition, it will be noted that the top of the working face of the hob tooth cuts the base circle beyond the point of

(*Continued on Page* 142)

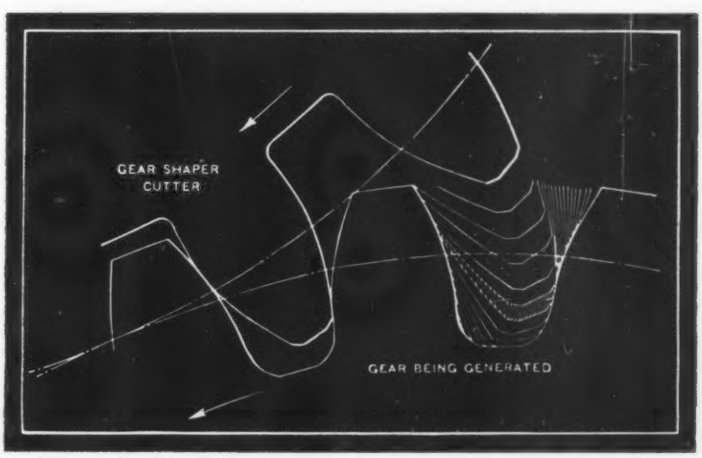


Fig. 6—Reciprocating cutter has same shape as a pinion of the system to be cut. Blank and cutter rotate in opposite directions

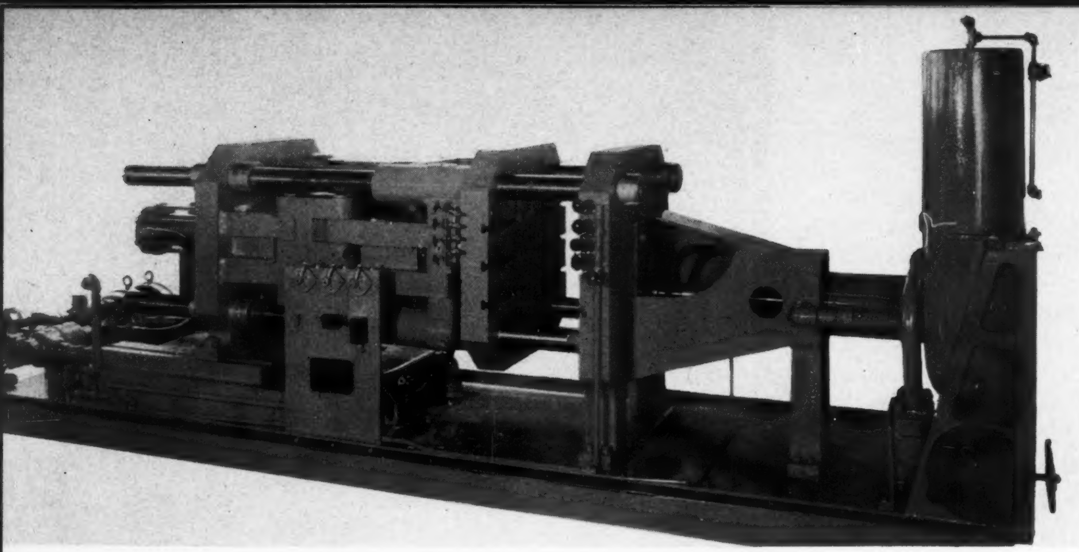


Fig. 1—Machine for die casting aluminum alloy for airplane parts, machine gun parts, fuse casings on shells and bomb tails

Valve

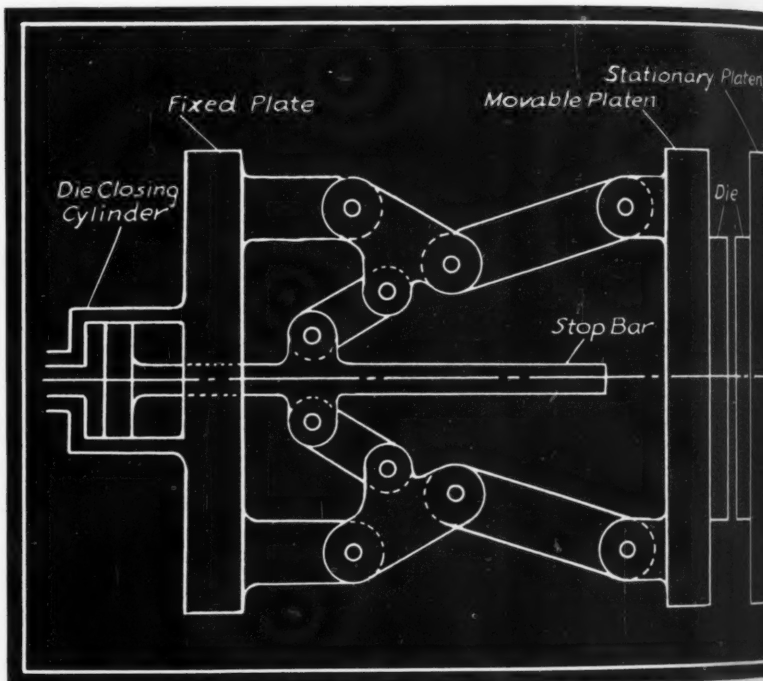


Fig. 2 — Above — Die-closing linkage operates fast and with high pressure. Stop bar minimizes stresses on closed linkage

Fig. 3 — Right — Die-closing end with linkage in open position

MANUALLY operated pilot valves, mechanically interlocked, actuate two four-way valves, hydraulically interlocked, to effect simple and efficient control of a die-casting machine. To produce a desirable product in this type of machine two high pressures are involved, one to hold the die together and the other to force the metal into the closed die. These operations are each controlled by their corresponding pilot and four-way valves.

Speed of operation and pressures involved demand substantial construction. For this reason the machine shown in *Fig.* 1 for die casting aluminum alloys is fabricated of heavy plate, flame cut and welded. Die platens are 5 to 6 inches thick and the linkage system for closing the dies 3½ to 4 inches thick, also flame cut from plate. Four adjustable tie bars for accommodating various dies run through the machine and are 4-inch diameter, SAE 1045.

Design of the linkage system schematically illustrated in *Fig.* 2 gives a closing pressure of approximately ¾-million pounds obtained from an hydraulic pressure system of about 300 pounds against the actuating mechanism. Arrangement of the linkage as shown in the illustration provides a fast-closing speed diminishing at the end of the stroke to build up a substantial closing pressure. Because the use of shoes or lugs in the linkage for over-center stops would build up excessive pressures and cause high shear stresses on the pins, a stop bar is employed as shown to close against the movable platen, thus providing positive closing without high stresses. Linkage in the open position is shown in *Fig.* 3.

Hardened steel bushings with extra wall thickness are used on the linkage system to prevent any distortion from the extreme pressures. Link pins are chrome-moly alloy having a hard surface and a soft core. The piston rods and plunger rods are high-carbon-chrome-moly steel to prevent

Simplifies Casting Machine

By J. J. Rose
Chief Engineer
G & N. Mfg. Co.

buckling or breaking from the hydraulic pressure. To provide bearing surfaces on the movable platen and crosshead, centrifugally cast bronze bushings are utilized. Cold-drawn seamless tubing having a wall thickness of one inch comprises the hydraulic cylinder for which the ends are flame cut from SAE 1045 plate.

Shown schematically in *Fig.* 4 is the injector mechanism for forcing molten aluminum alloy into the closed mold. The injector is shown in the forward position after having been used to "bump" the gate to loosen the

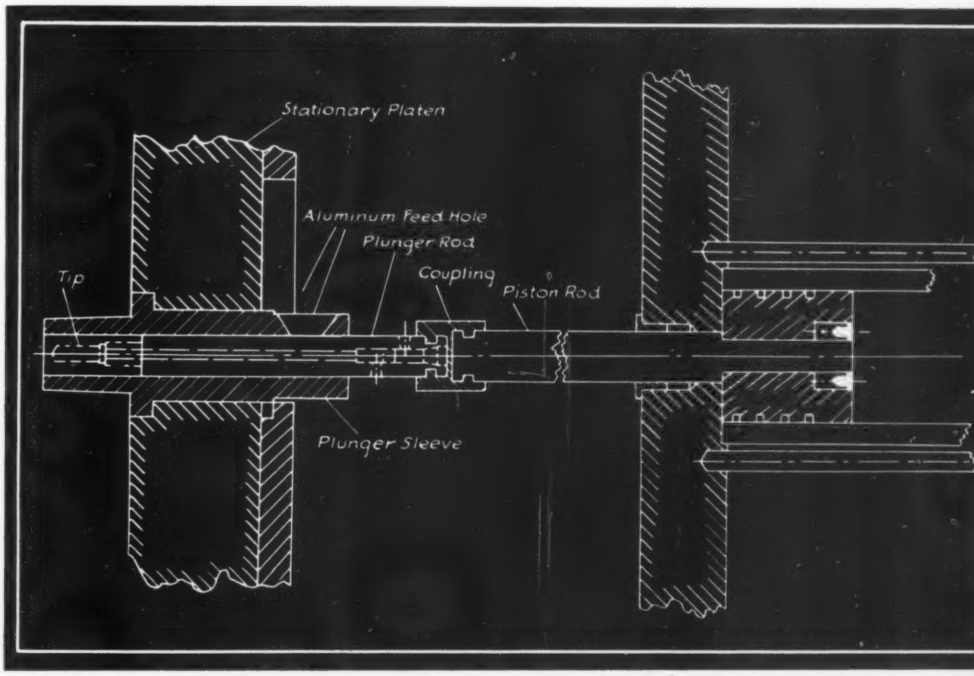
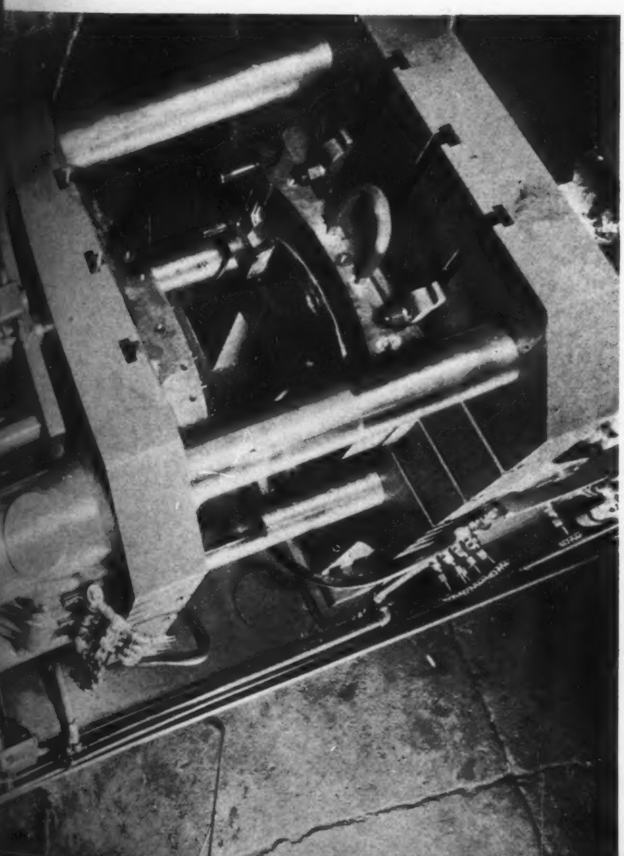


Fig. 4—Above—Injector unit showing feed port, method of watercooling and nitralloy tip



solidified casting. After return of the plunger rod molten aluminum is fed in the port hole shown and injected into the casting chamber by the forward movement of the plunger rod. Because aluminum has an affinity for iron the plunger tip and plunger sleeve are nitralloy. Plunger rod is watercooled by copper tubing and nitralloy tip serves as a water jacket for maximum cooling effect on end of plunger. Plunger rod, piston rod and piston are high-carbon-chrome-moly steel and the hydraulic cylinder is cold-drawn seamless tubing similar to die closing cylinder. Tubing has a wall thickness of one inch.

Because the machine requires constant attention for loading and operating, manual controls, *Fig.* 5, are indicated. A hydraulic pump of the two-stage type furnishes pressure for both the die-closing and shot mechanisms. The schematic diagram in *Fig.* 6 shows the primary hydraulic system with a pressure accumulator. For simplicity the pilot control system is not shown. This pilot is manually operated from the valve levers shown in *Fig.* 5

which position the four-way valves for the die-closing and injector cylinders. The valve controlling the shot piston has five ports and is connected to the accumulator bottle. Accumulator has an inert atmosphere of nitrogen and assures a uniform pressure throughout the shot-piston stroke. This nitrogen cushion also intensifies the speed of the shot.

To prevent reverse flow in pressure line during the injection cycle, a check valve is used in the main pressure line. The hydraulic system is all-welded pipe construction wherever possible to reduce friction and to give greater efficiency to the machine.

Hot metal end of the machine may be converted readily into a zinc, tin or lead machine by replacing the end assembly with a shot cylinder and metal pot unit. For zinc, tin or lead alloys interlocks are provided to protect the operator against danger of being burned with molten metal should he operate the wrong valve. For instance, in *Fig.* 5 the mechanical interlock shown on the valve prevents operation of the shot piston unless the dies are closed. Another interlock is also provided and is shown in *Fig.* 3 above the linkage mechanism. A hydraulic valve is in series with shot-piston valve and will not allow the latter to be operated unless the die is closed, depressing a plunger on the safety valve by means of a pin on the die-closing mechanism.

An additional check valve is utilized when casting zinc, tin or lead. It provides a return vacuum action of the shot plunger, permitting instant opening of the die mechanism without any danger of pucking.

Shot cylinder is extra heavy with flame-cut alloy steel heads, the lower one of which is water-cooled. Gooseneck and shot-cylinder support are flame cut and welded plates, adjustable on three-inch threaded bars. Gooseneck and metal pot are cast low-chrome-nickel steel for corrosion and heat resistance and are removable without disturbing support. Nozzle is also cast of the same material. Burner for heating pot has a three-inch blower, proportionate mixer and temperature-control motor, all mounted on the base to provide accessibility for maintenance.

Careful alignment, rigidity and elimination of bolted construction contribute toward maintaining high pressure on dies during injection of molten metal. Thus it is possible to produce a satisfactorily dense casting with a minimum of flash which requires trimming.

For small simple castings without cores or complex parts and using automatic ejection it is possible to operate the machine at a speed of 600 cycles per hour. For larger and more complex castings the rate may go as low as 15 cycles per hour.

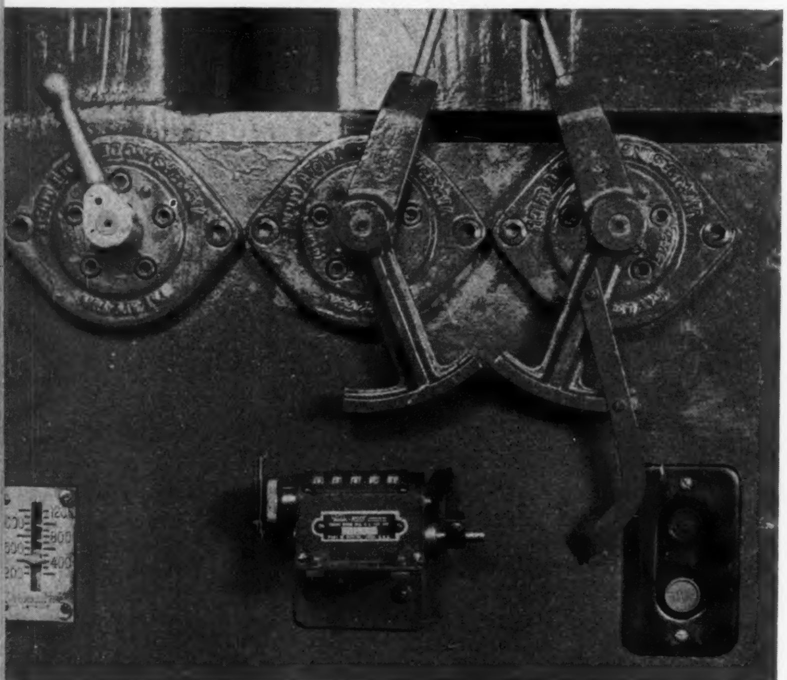


Fig. 5 — Above — Pilot control valves have interlocks to prevent operation in wrong order

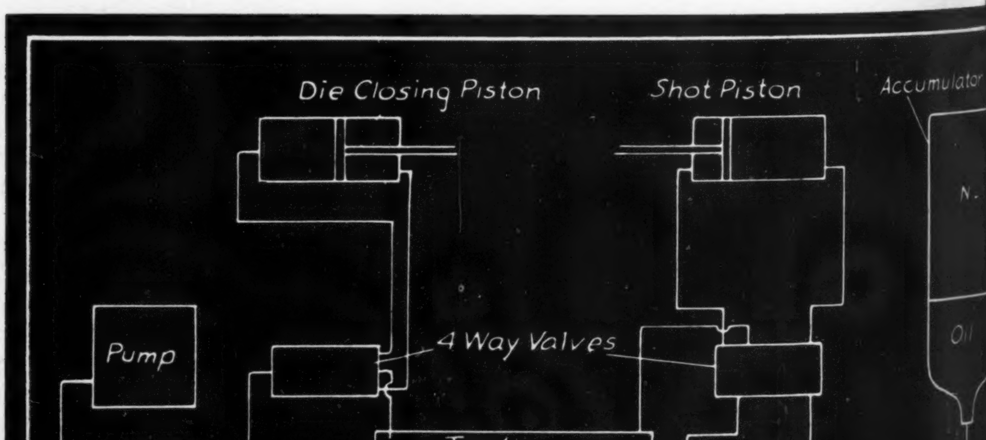


Fig. 6 — Right — Hydraulic system for die-closing and injection operations. Position of four-way valves is controlled by pilot system not shown

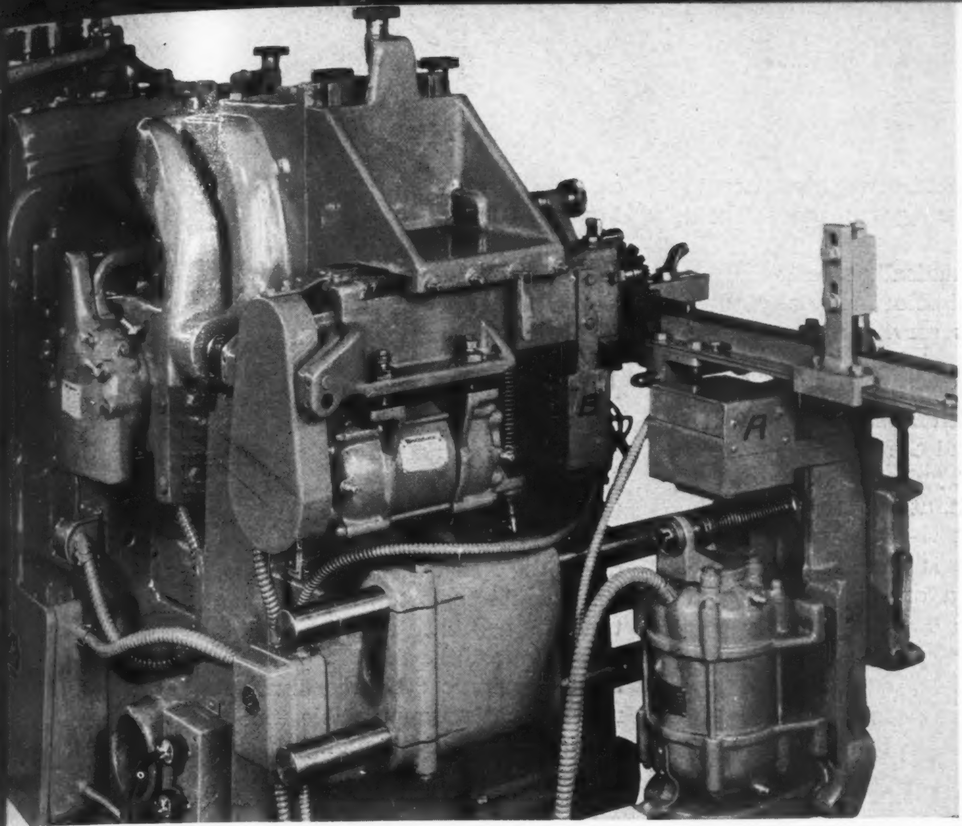


Fig. 1 — Solenoids accurately control cutters on woodworking machine operating at 120 feet per minute with 40 cutting operations

Sequence Operations Speeded with Solenoids

By D. K. Frost

Electrical Engineer
Mattison Machine Works

IN PERFORMING or controlling machine operations solenoids usually are remotely controlled by the closing of pushbutton contacts or a pilot device such as a limit switch, float switch, thermostat, timer or pressure switch. Solenoids are, therefore, definitely tied in with so-called sequence or automatic cycle controls now being used extensively to speed production in aircraft and other war equipment plants.

It is impossible to design solenoids to meet all requirements. But, if a problem is approached with the realization that it is not only important but necessary to select solenoids whose specifications exactly fit the load requirements, there should be no difficulty encountered as far as adaptability is concerned.

There are other considerations, however, which complicate the problem. These lie in the fact that solenoids often do not enjoy immunity from surrounding conditions such as oily vapor, moisture, metallic dust and mechanical injury. Solenoids used in magnetic switches and relays are commonly mounted in enclosing cases or panels with semi-dust-tight enclosures and have attention from maintenance crews. On the other hand, solenoids for remote control of valves, for instance, must be placed close to the part they operate. In these locations they are subject to adverse conditions and unless properly protected may deteriorate and cause trouble. Most solenoids, however, are wound with wire having impregnated and baked insulation which is not affected by oily vapor. As a factor of safety, solenoids for machine tool ap-

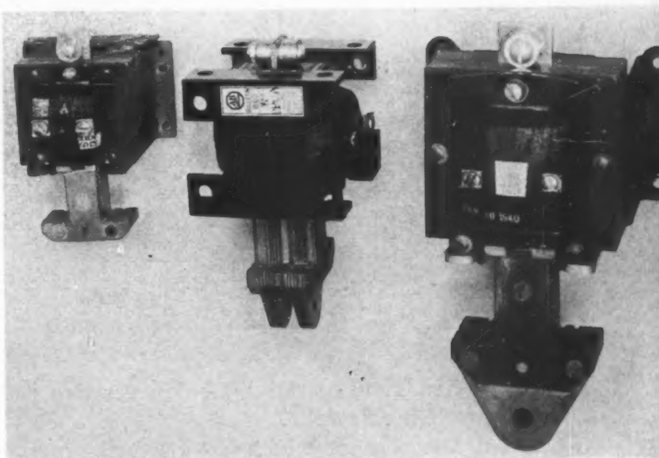


Fig. 2—Typical solenoid units ranging in size from .6-pound pull to 5 pounds

plications should be capable of being immersed in oil without causing any damage to the solenoid unit.

Solenoids should be mechanically strong and all parts subject to wear should be of material capable of withstanding continuous service after millions of operating strokes. Magnet laminations should be supported by heavy side pieces and firmly riveted with solid steel rivets, with a sufficiently large factor of safety to withstand the continuous hammer blows of the plunger. Typical units are shown in *Fig.* 2. The smallest unit, at the left, is capable of a 6-pound push or pull at ¾-inch stroke, whereas the unit in the middle is capable of a one-pound pull or push with a one-inch stroke and the solenoid at the right delivers five pounds with a one-inch stroke.

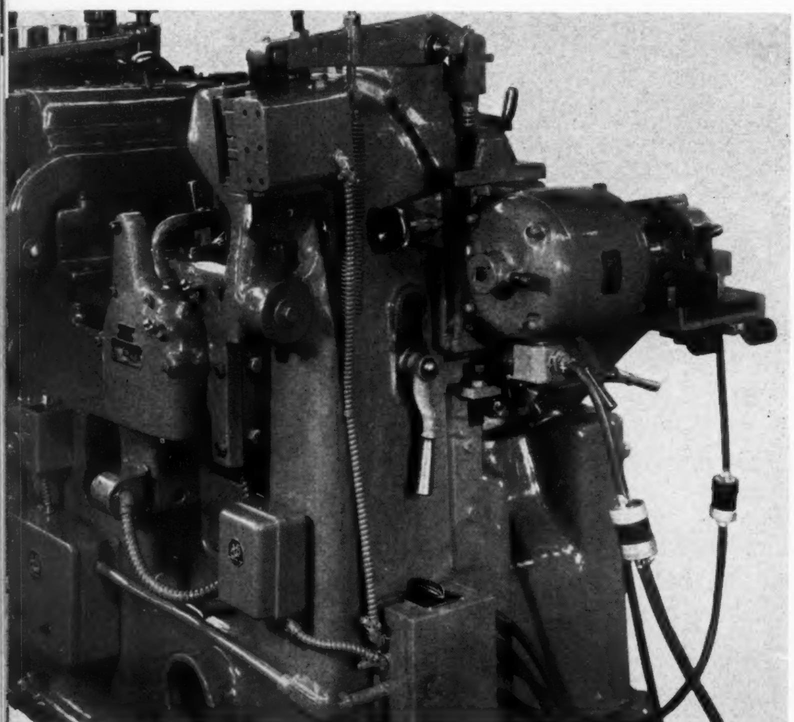


Fig. 3—Heavy-duty solenoid is required where motor as well as cutter is moved

The plunger which travels through the center of the coil is generally an inverted *T* construction similar to a hammer head. It is provided with a hole to which a linkage may be connected for pulling or pushing action. This linkage and the hole through the hammer head in the pull type or the hole for the connections to the upper part of the plunger for the push type have been sources of trouble from wear or from binding due to misalignment. More favorable results can be obtained by obviating this linkage and using push-type solenoids with a retroacting spring on the device to return the plunger to its original position after the power stroke has been completed. Also, by decreasing the length of stroke by fifty per cent, the number of operations per minute may be increased as much as three to four times without overloading the unit.

In modern control, speedy operation of solenoids as well as relays and switches is important. Where any slight delay in dropping out might result in faulty operation, an additional pull out magnet may be necessary. Solenoids must not only operate quickly, but also consistently so that when the cycle is once set up it will repeat time after time with unfailing accuracy. For instance, in passing from rapid approach to cutting speeds, if the fast traverse speed is high and the cutting speed slow it is apparent at once that the point of cutting down the speed must be accurate since too much time would be lost with any appreciable space or time in approaching the work at slow cutting speed. It will be found that the space allowed on most automatic cycle machines is so close in this respect that no appreciable time is lost; however, special means of obtaining this accuracy must be supplied in each case.

Capable of Fast Operation

An extremely fast solenoid-controlled machine is shown in *Fig.* 1. Window and screen sash is fed into this machine by a hopper at a rate of 120 feet per minute. There are two cutters, one vertical and one horizontal which operate on each piece as it passes. At 120 feet per minute there are about 40 cycles or pieces per minute. The cut made is about ⅜-inch wide by ½-inch deep and 3 to 4 inches long and must be in the same location on each piece. This is accomplished on runs of many thousands of feet with no appreciable variation.

When the feed rate was stepped up to 180 feet per minute or at the rate of a cut each second, some slight variation was found. On still faster numbers of operations the variation was still more apparent and unquestionably was caused by the fact that in alternating current the solenoid may take 5 cycles to close on one operation and 5⅛ cycles on the next as the impulse from the pilot limit switch might come at the moment when the voltage was at or near zero in the alternating-current cycle.

In *Fig.* 3 is shown a similar machine but where the motor as well as the cutter is moved up and down instead of the spindle and cutter only. With the increased inertia it is impossible to obtain high rapidity of operation from the large solenoid lifting the weight of the motor. Operation of this machine, however, need not be more than 50 or 60 feet per minute. Where extreme rapidity of operation is required it is necessary to reduce the weight and inertia to the smallest allowable amount for dependable operation.

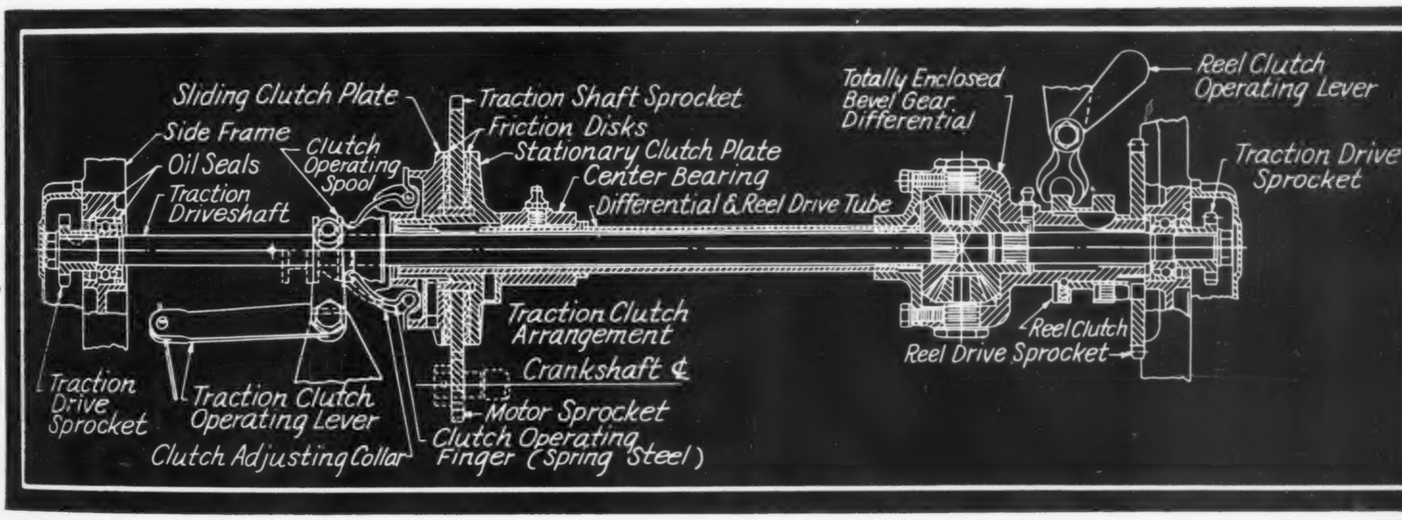
By carefully designing applications and recognizing the limitations of solenoids it is possible to obtain satisfactory and trouble-free operation, as with other types of electrical equipment.

Mower Drive Is Designed for Minimum Maintenance

By Einar Jacobsen
Vice President
Jacobsen Mfg. Co.

DEVOTED to drives and controls developed by machine designers in various fields, the accompanying articles are representative of the ingenuity exercised by engineers in these difficult times. Methods of incorporating commercial drive and control elements into machines, as discussed in the several articles, should be of broad interest to the profession

MACHINES and mechanical equipment such as the power lawn mower discussed herein, which are by no means certain to receive proper care and maintenance, require a maximum of attention to sturdiness and reliability in their design. These features combined with relatively high-speed operation of some of the components



make the drive problem involved an intriguing one.

This drive features the arrangement of two separate drives, each with its own independently operated clutch, about a common driveshaft and, in addition, having a fully enclosed bevel gear differential for the traction drive.

Power is derived from a one-horsepower two-cycle horizontal engine driving, by chain, the traction shaft sprocket. This sprocket is also the driving plate of the traction clutch, the power being transmitted through a friction disk on each side of the sprocket. The clutch itself is actuated by means of a spool engaging two spring-steel fingers or levers, providing a resilient engagement of the sliding clutch plate. The stationary clutch plate is keyed to a drive tube which transmits the torque to the differential case and thence through the bevel gears to the traction shaft, which rotates inside the tube. This shaft is divided into two parts extending to each side of the mower from the differential. The farther end of each is mounted on enclosed ball bearings in the mower side-frames. The final drive to the traction wheels is again by means of sprockets and chains, all-enclosed and operating in an oilbath.

Mounted integrally with the differential case is the reel drive-sprocket and clutch. The sprocket normally floats on a bronze bearing on the differential case when the clutch is disengaged. A sliding collar, keyed to the differential case, and having driving pins to engage in holes in the reel sprocket, provides the means of transmitting the drive to the sprocket and thence by chain to the

cutter reel. The schematic diagram indicates the general arrangement of sprockets and chains used in the entire drive.

Spring-finger engagement of the traction clutch not only provides resilience, but eliminates any need for critical adjustment as would be the case with solid levers. The collar supporting these fingers is adjustable for take-up.

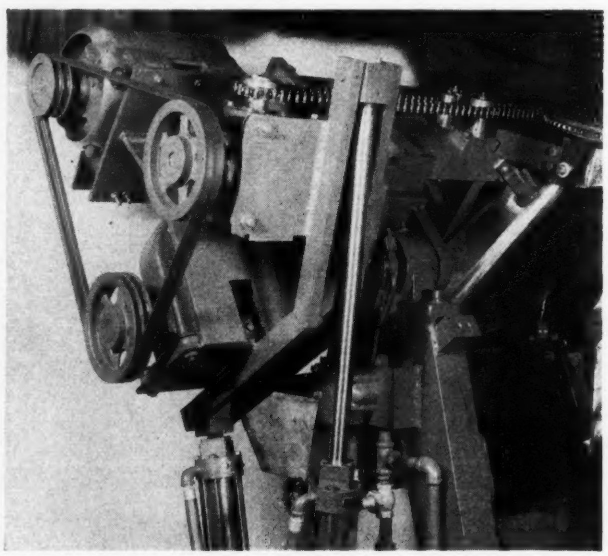
Utilizes Standard Drive Units

John V. Davis

Chief Engineer
The Udylite Corp.

FOR broad diversification of its problems the design of the drives and controls for the automatic electroplating equipment illustrated has few equals. Not only are the usual problems of prime mover, power transmission, bearings, etc., to be coped with but also the chemical and electrochemical processes pose many difficulties for the engineer. Not least among these are corrosive attack and the isolation of stray electric currents.

As is evident from the illustration the drive design is exceedingly straight-forward. Necessity for tilting the banks of perforated chambers into and out of the solution dictated the mounting of the rack or frame on horizontally pivoted joints. Two air or hydraulic cylinders are then used to provide the necessary power. Control and timing is achieved by solenoid valves, operated through a timer relay.



Rotation of the perforated chambers about their own axis is also necessary in order to induce uniform exposure to the electrolyte as well as to expedite draining when lifted from the bath. This is accomplished simply and effectively by mounting an electric motor directly on the oscillating frame. Thence power is transmitted via V-belts through a pair of gear reducers to two roller chain sprockets, one for each bank of chambers. Around the periphery of the neck of each chamber is a sprocket in permanent engagement with, and driven by, the roller chain. As may be observed from the illustration, idler sprockets are interposed at intervals along the chain to maintain an adequate angle of contact between the chain and the chamber sprockets.

Possibility of pitting of the bearings as a result of stray electric currents must be eliminated. First, every effort has been made to anticipate possible current paths and direct them away from bearings. Because of the magnitude of the currents involved as well as the reverse polarity of the cleaning solutions, this is not always feasible. Therefore, as large bearing areas as possible are selected in order to reduce to a minimum the current density across any bearing surface.

Crusher Poses Severe Drive Problems

By Homer W. Riley

Chief Engineer
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UNLIKE some crushing problems, coal crushing demands "size" production, i.e., the breaking of the feed to as near a certain size as it is possible. These sizes vary greatly from job to job, dependent on market demand. Since coal may be extremely hard or soft, tough or brittle, and possessed of varying degrees of friability, the design of the crusher used is all important. This is especially true when it is known that "fines," or coal that is too small to sell, and "oversize," that which is larger than specified, determine to a great extent the profit accruing

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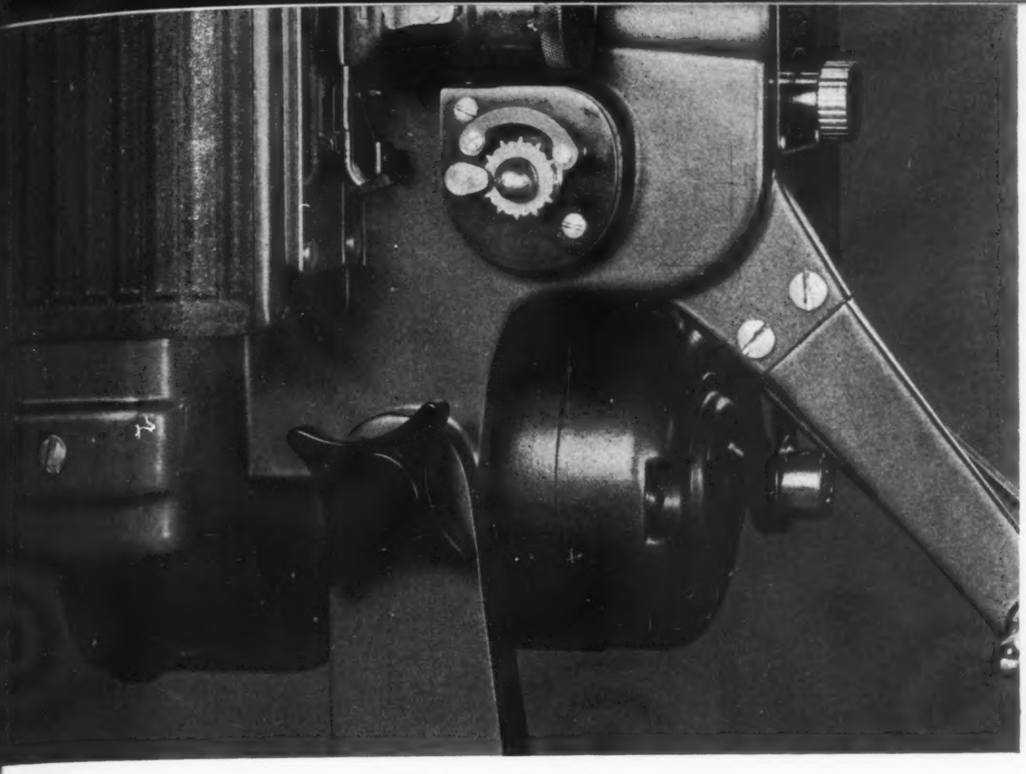


Fig. 1 — Appearance, compactness and light weight are essential to portable machines like projectors

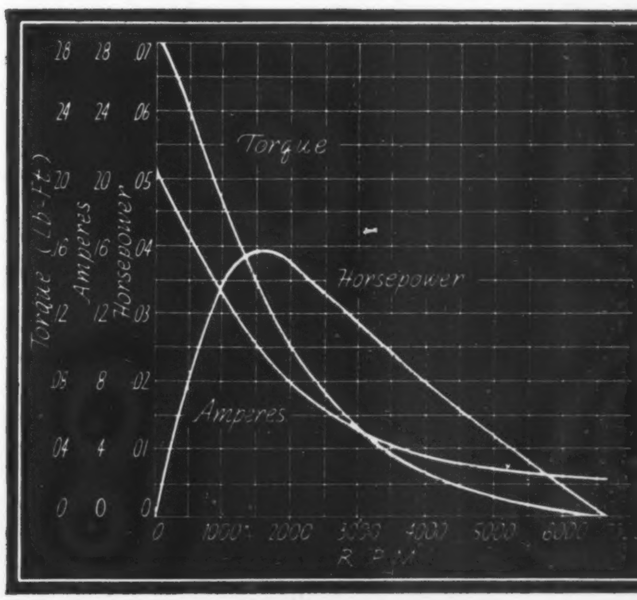
Fitting Miniature Motors To the Job

By Vaughn H. Hardy

Delco Appliance Division

SMALL fractional-horsepower motors often must be designed for a particular application. Unlike the larger size motors, there are no set NEMA dimensions and, more often than not, the dimensions of the motor must fit a definite space. This means that for such applications the motor is really tailor-made.

Thus, before specifying a motor, the designer must take into consideration all the facts involved. Some of the more pertinent of these are power and speed required, size and weight limitations, ambient surroundings, desired efficiency,



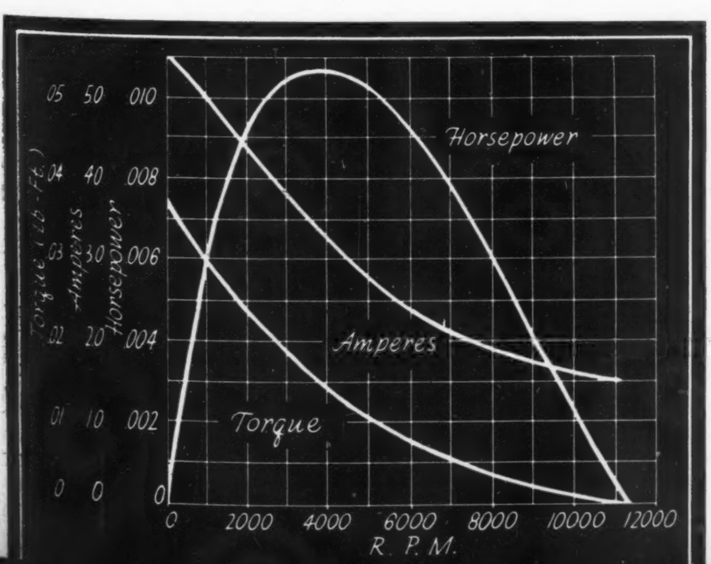


Fig. 2—Left—Characteristics of series-wound, six-volt, direct-current motor for individual defrosters

Fig. 3—Above—Series-wound, six-volt, direct-current blower motor for underseat heater develops high starting torque

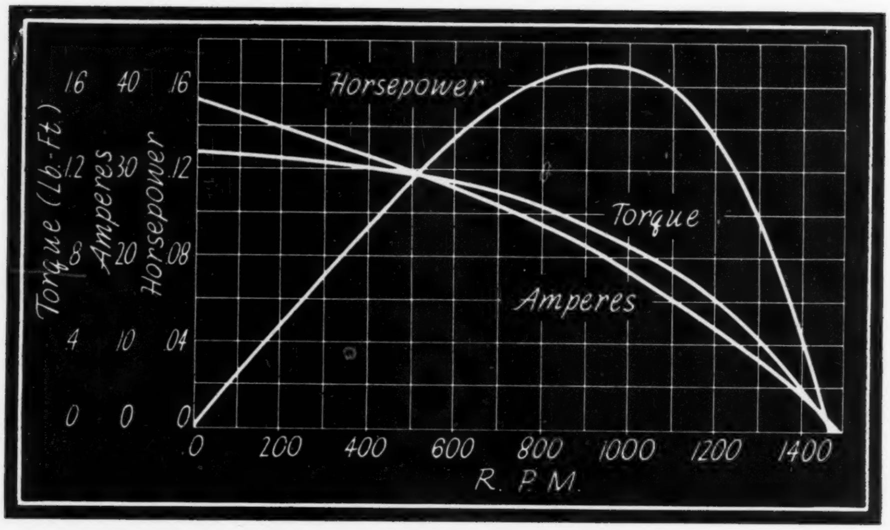
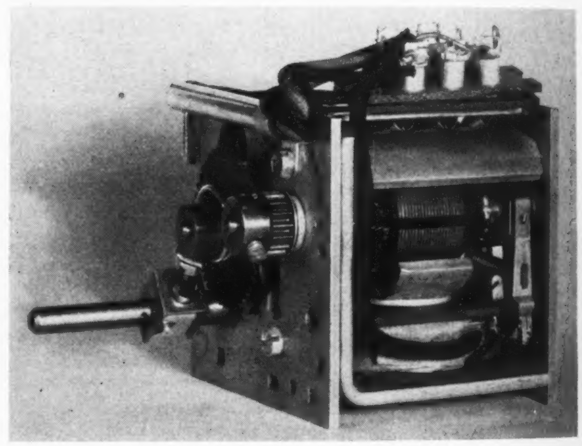


Fig. 4 — Left — Shunt-wound twelve - volt, direct - current motor for transcontinental bus air conditioning units

Fig. 5—Below—Special built-in series motor applied to radio tuning unit



and approximate yearly volume. In discussing applications such as *Fig.* 1, later in the article, the importance of these various items will be brought out.

Probably the largest volume of all small motors has gone into automobile defrosters and car heaters. Because of this fact and the necessity for a low-cost motor, a particular type of construction was evolved. Each motor is purposely designed to give the required power at low cost. All parts, including cases, must be inexpensive and are, therefore, usually stampings and screw mamachine parts. The motors must be easy to assemble because they are put together on a high volume assembly line by specialized machinery. In making this type of motor, the engineer is not hampered in his original design by tool costs, as these will be spread over a large volume. However, once the motor is in production, because of the large investment in specialized machinery and tools, only minor changes can be made. Characteristics of these motors are shown in *Figs.* 2 and 3.

There are also many limitations on the design: Size of the motor must be kept a minimum, especially if for the dash heater and defroster; motor noise also must be as low as possible; ambient temperature is important since temperatures from —30 to 180 degrees Fahr. are often encountered.

Because of the desire for speed regulation by resistors, as well as high starting torque for cold weather, these motors are limited to series wound. With the advent of the underseat heater and the fresh air ventilators where the motor is exposed to the weather and dirt, special protection is provided.

Another application which has become important is the window raising and lowering motor. This same type of motor is also used on the convertible top lift mechanism. The motor delivers high torque and power for its size but its runs are intermittent and of short duration. Thus, by pushing the power output far beyond the ordinary, the motor develops about seven times its normal continuous rating.

An example of a different type of application is a bus ventilating fan motor used for air conditioning purposes. In this instance the question of long life becomes paramount as the fan motor is in use continuously. These motors are usually inaccessible. Thus, all efforts are bent to insure reliability and long life. In modern quiet buses noise level is becoming an important factor also. This requires close bearing alignment, quiet brushes and careful balance. Characteristics of the motor are shown in *Fig.* 4.

Automobile radios require a special type of motor to operate the tuning mechanism. Because the motor is enclosed in the radio housing, it can be of open-type construction. Since space and cost are also of prime importance, the motor is made up of stamped brackets for the frame with a solid pole piece. In this motor *Fig.* 5, the frames are staked together and the pole piece is resistance-welded in place.

Another small motor application is in moving picture projectors for amateur use. Since variable speed is desirable, as well as universal operation,

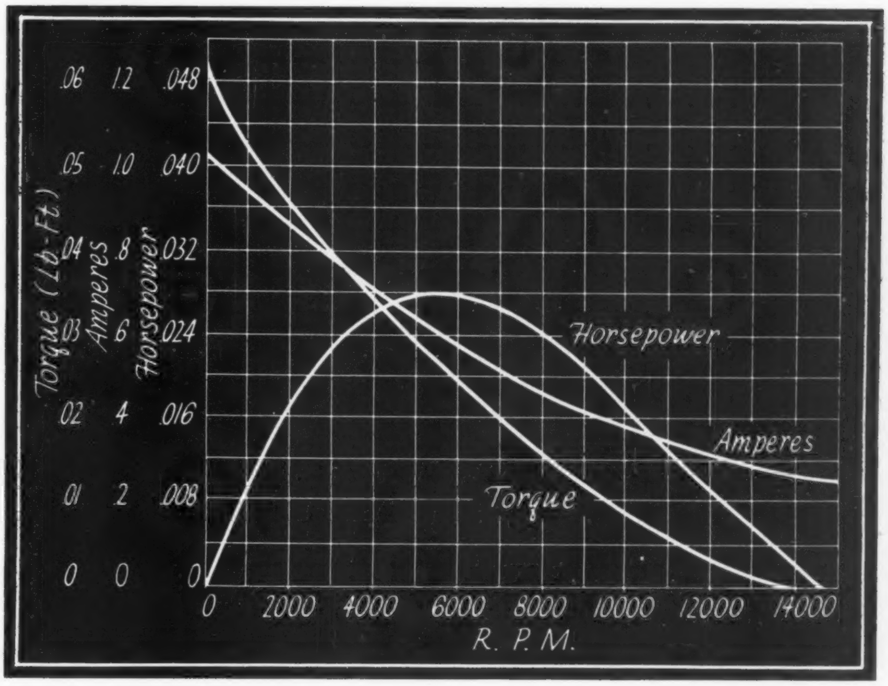
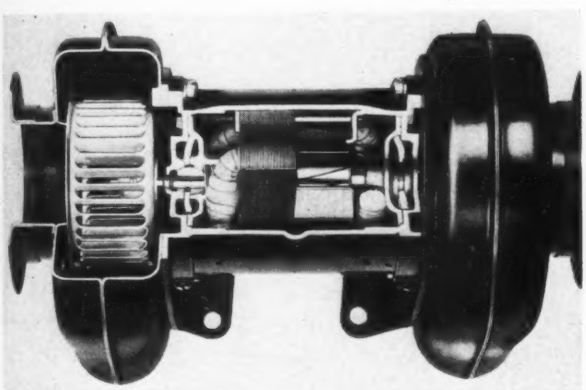


Fig. 6—Right—Characteristics of a small series-wound universal motor

Fig. 7—Below—Motor applied to a twin blower unit



a series motor is necessarily used. In cases where the motor is exposed appearance becomes important and a die-cast case is used as in *Fig.* 1. In other instances the motor is built in as part of the mechanism and the motor manufacturer supplies only the parts. Motor size and weight are important because the projectors must be light and compact for easy portability. Noise level of the motor also must be low. Characteristics of a small universal motor are shown in *Fig.* 6, for torque, horsepower and amperes plotted against revolutions per minute.

Small gear-reduction motors have such a variety of applications that they cannot be fully covered here. If any volume is involved, they usually require a special design to fit the job in hand. If only a few are required, a more or less standard unit usually can be found to do the work. This would be uneconomical for large production items as a carefully designed geared motor would cut the cost and improve the performance. A special job is also necessary where there is a limit on size and weight or where good appearance is essential. Whether the gearing is an enclosed or open type depends on the conditions to be met as well as the type of gears and the type of motor.

Small blowers as in *Fig.* 7 are another application where these motors are used extensively. Both alternating and direct current are employed, so that these motors must be made interchangeable. For alternating current a small two-pole, shaded-pole motor is used, while for direct current a series motor is employed. These motors have extra-large oil reservoirs, as long life is important. For specially long life, with low wattage consumption, condenser motors with sealed ball bearings are necessary. These are used on small domestic oil heaters where the motor is required to run almost continuously.

High-speed hand tools present additional prob-

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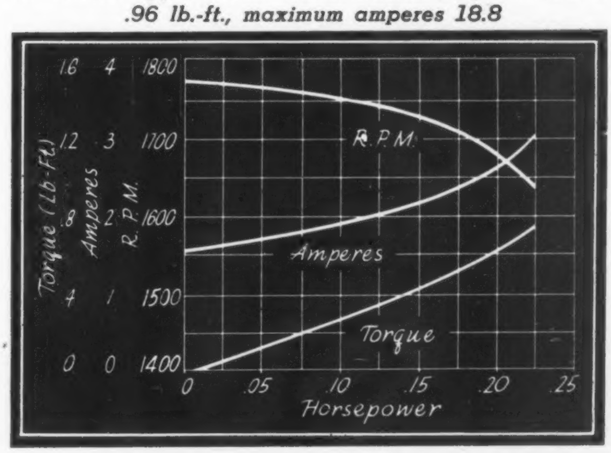
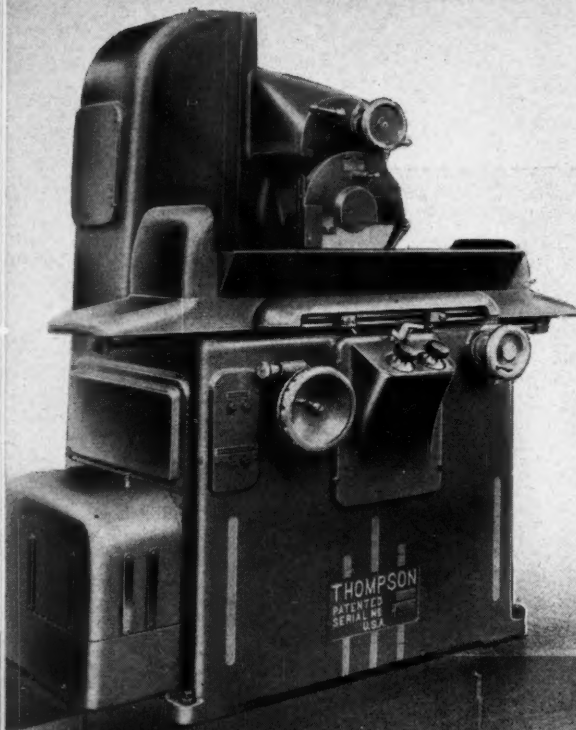
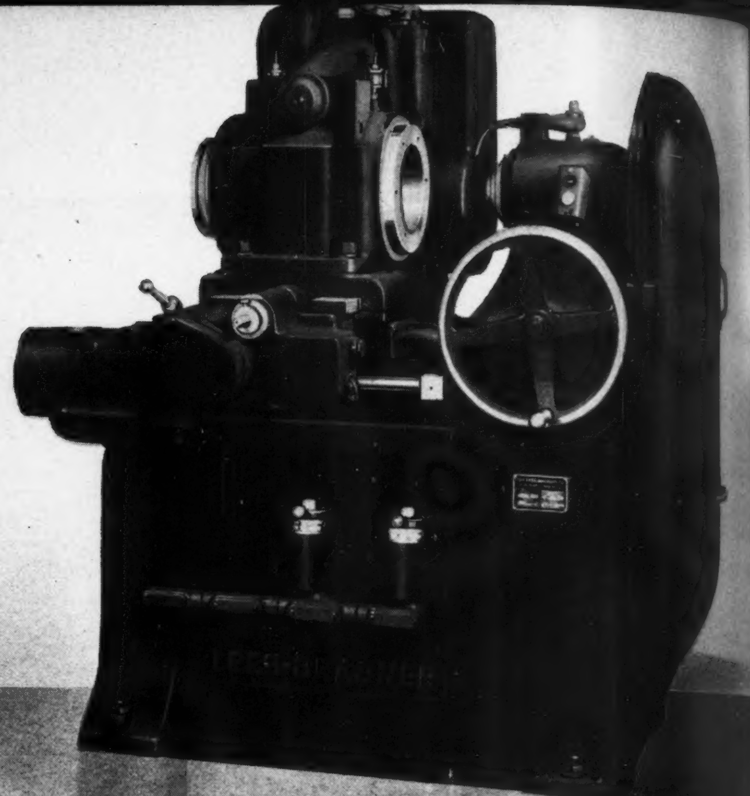
Fig. 8—Split-phase motor characteristics for domestic oil burner application. Locked rotor torque is .88 to .96 lb.-ft., maximum amperes 18.8





Above—Table and wheelhead movements of Thompson surface grinder may be accomplished either hydraulically or manually. Combination of antifriction nut and screw drive for table feed insures positioning with a minimum effort. Telescopic pipes are used in the hydraulic circuit



Above—Workhead of Lees-Bradner thread miller is driven by an electric motor mounted directly on the head, eliminating long driveshaft. For generation of registered threads a solenoid-operated disk-type brake is employed. Pickoff gears provide variable speed for work spindle; cutter spindle speeds are varied by changing V-belt sheaves

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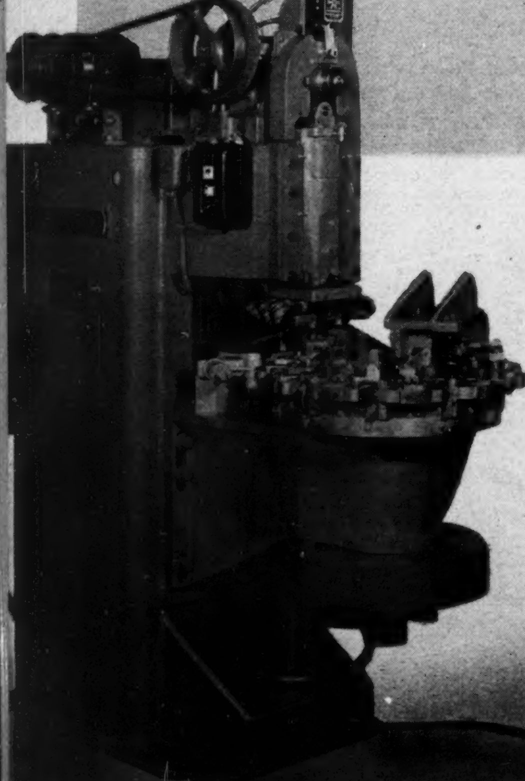
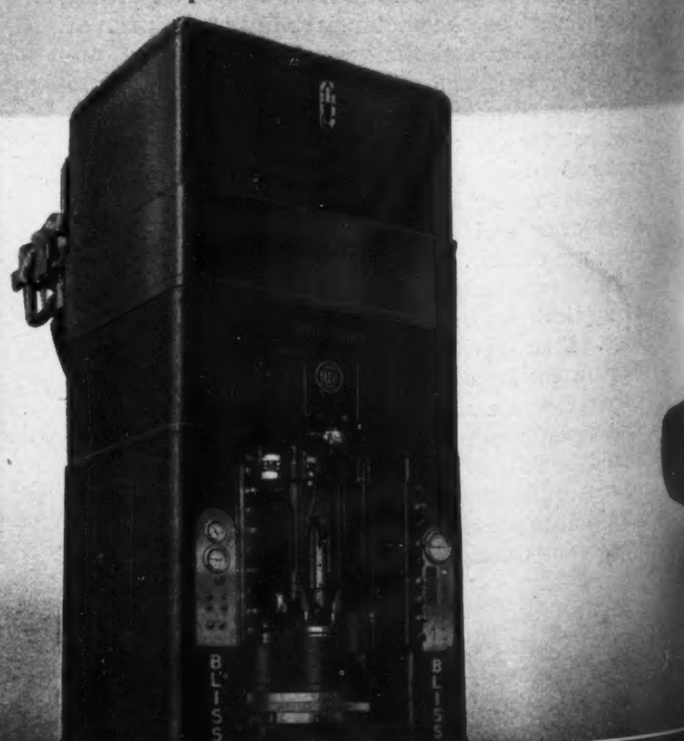
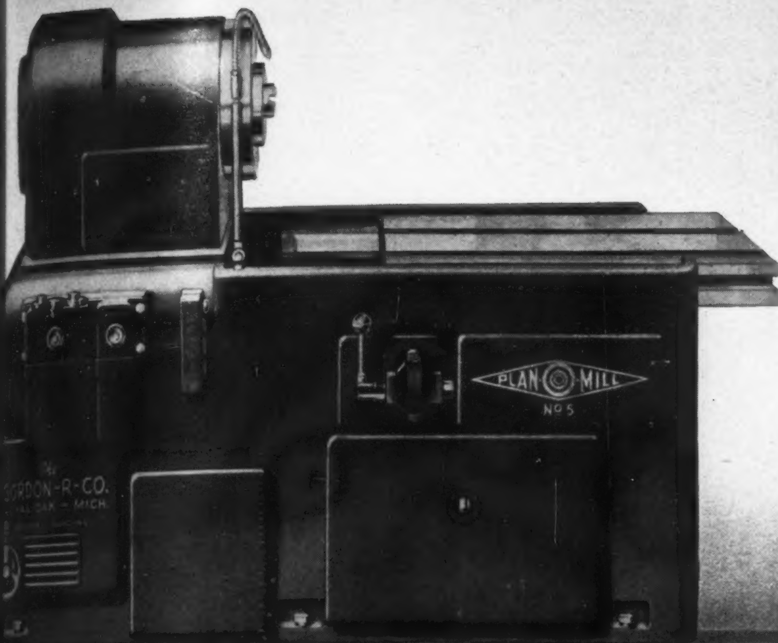
Below—Equipped with a ten-station automatic dial feed, Thompson-Gibb automatic shell welder is powered from a seventy-five kilovolt-ampere transformer. Indexing table is linked directly to the main drive and is synchronized with the action of the welding head. Welding current is controlled electronically

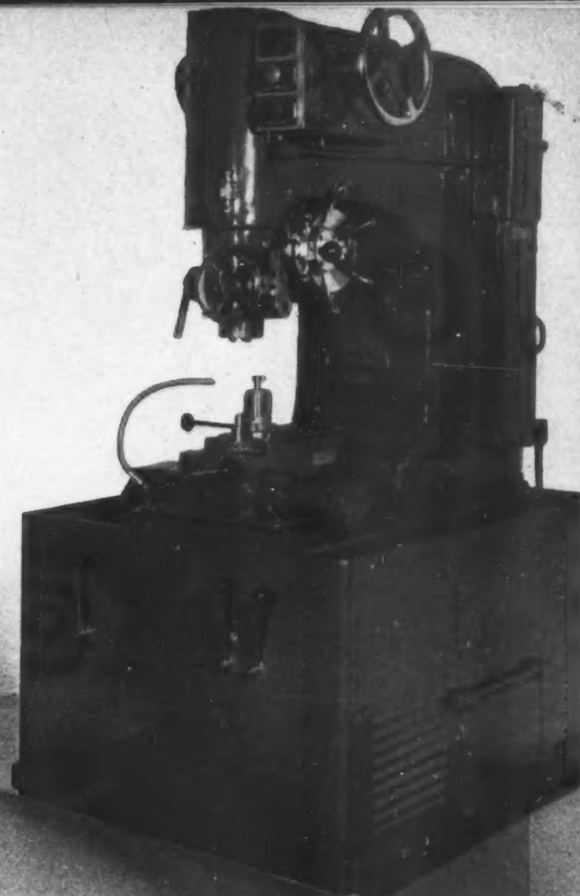
Below—Press of 2000-ton capacity for heading cartridge cases is designed to be driven either hydraulically or by a mechanical knuckle joint linkage. Bliss press is self-contained, being operated by a 75-horsepower main drive motor and a $7\frac{1}{2}$-horsepower ejector drive motor. Equipped with a two-station lower dial pneumatically operated, hydraulic dashpots are used to eliminate shock







—Flexibility of hydraulic operation applied to planetary mill- characterizes the Gordon-R machine. Hydraulic motors driving spindle and quills provide an infinitely variable selection of and speeds by merely adjusting dials. Vane-type hydraulic ment is used in this application, as well as in the operation of work table. Built-in selector switch permits automatic or manual operation at operator's discretion



Above—Electric control of full hydraulic drive of semi-automatic Snyder milling machine makes possible the roughing and finishing of radial airplane engine master-connecting rods by one unskilled operator. Power transmission to quill-type spindle is a wormwheel unit

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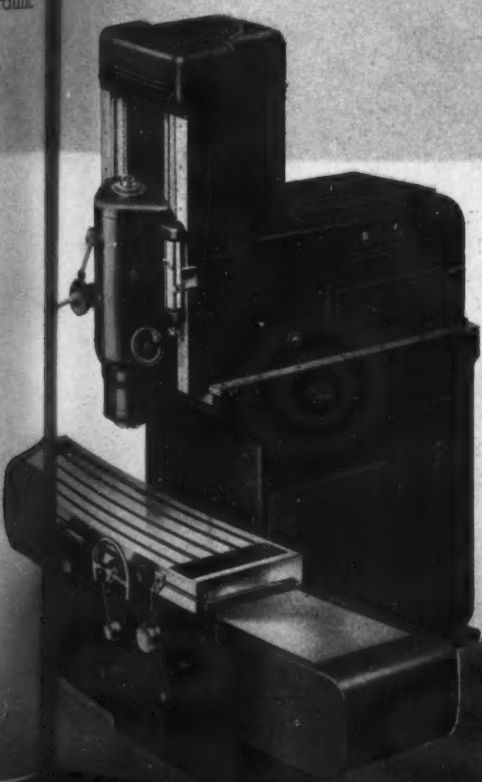
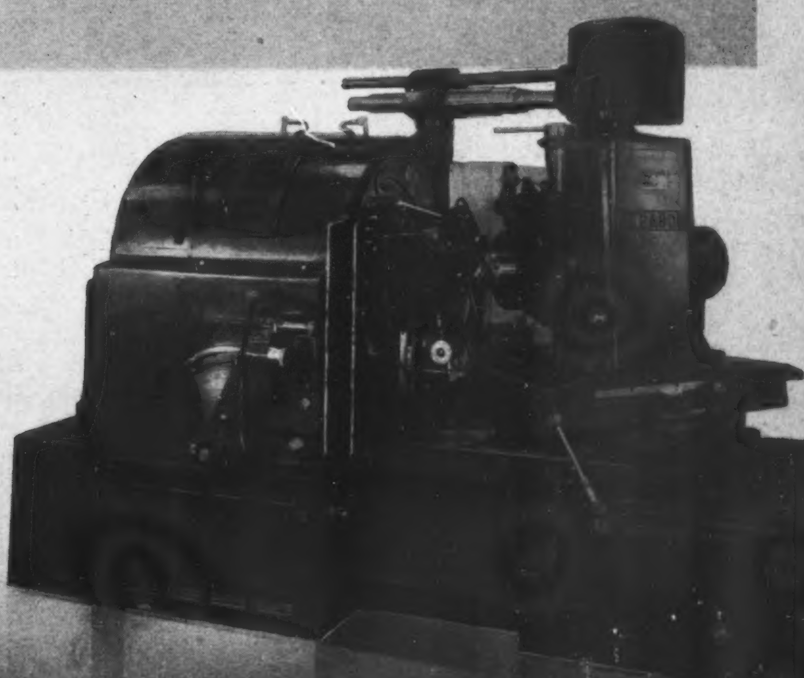
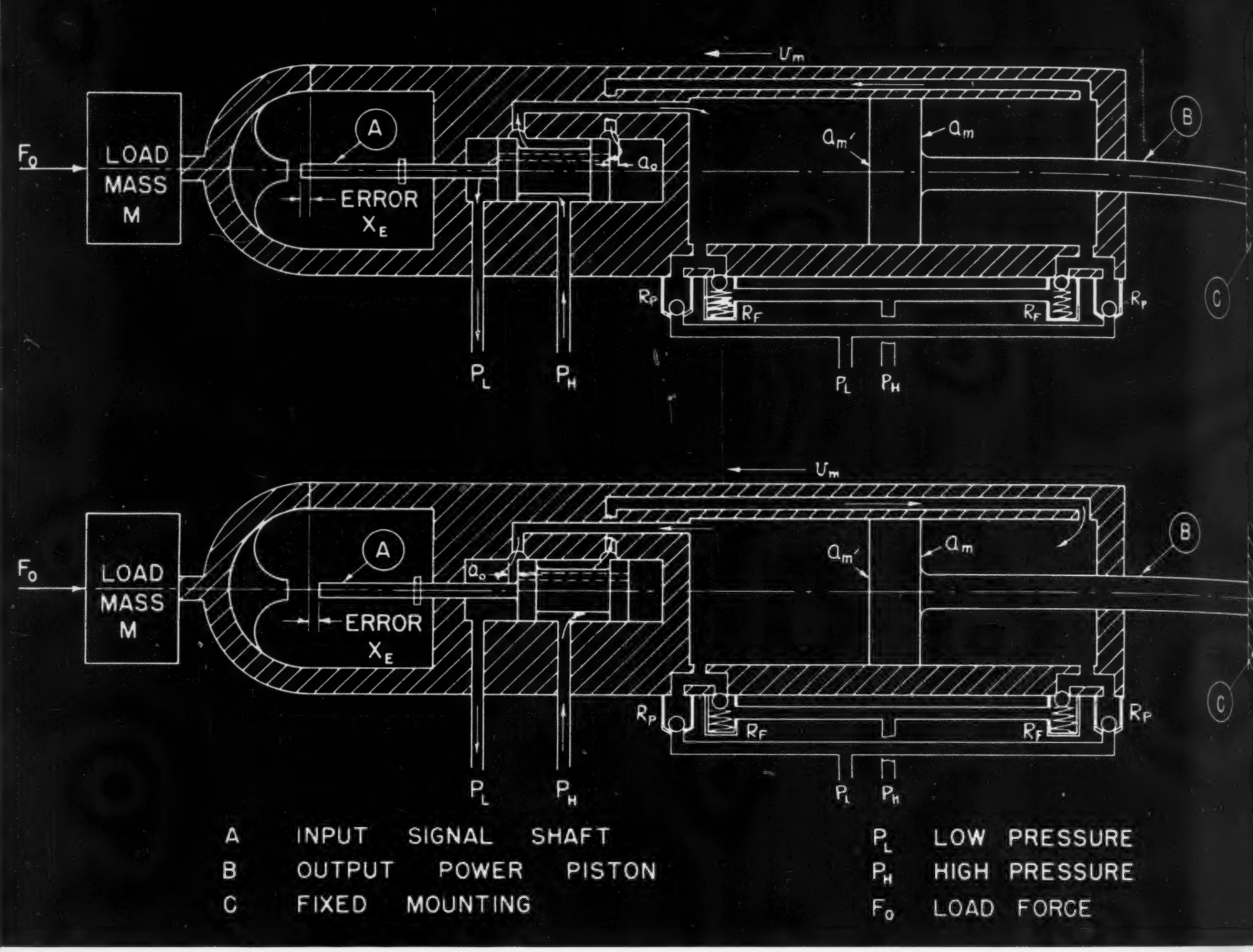
(For see Pages 232-234)

w—Spindle drive gear box of Reed-Prentice vertical milling ma- e is mounted on cross slide and provides nine speed changes ted by only one lever. A second lever, varying the displace- of a hydraulic motor, controls the feed and rapid traverse of the work table. Cross slide is also hydraulically operated

Below—Generating motion of Gleason hypoid gear grinder combines continuous rotation of the work spindle with a reciprocating motion of the wheel-carrying cradle. Hydraulic movement of sliding base to and from cutting position provides speed and ease in changing the work







Applying Hydraulic Servo Circuits in Machines

By Christian E. Grosser

IN AN earlier article on hydraulic servo design in the January, 1941 issue, the fundamental principles underlying servo operation in general, and the application of hydraulic equipment to servos, were discussed briefly. In addition a simple pressure-controlled hydraulic servo type was described, a method proposed for predicting the response of a given design to signal, and recommendations made for minimizing errors of lag and oscillation.

This article presents a specific application of such a pressure-controlled servo. Reasonable proportions and load conditions are assumed and detailed calculations presented which follow the aforementioned method of analysis of response. Quantitative results are shown in curve form for a ready appreciation of the response characteristic, together with the variations in hydraulic pressures and velocities occurring in the circuit.

The application considered is a steering booster for use on heavy vehicles where the forces necessary to move the conventional mechanical steer-

Fig. 1—Top—Two positions of signal shaft, corresponding to displacements of steering wheel, illustrate flow of oil through control valve

ing gear are so large as to necessitate great effort on the part of the operator.

The booster acts to amplify a small effort so that precise control is obtained. The servo, or booster, responds instantaneously, by moving the load against large friction forces, in correspondence with the operator-controlled signal. Exactness of the response to the control movement in a typical design of booster will be the chief subject of the discussion.

Fig. 1 shows in diagrammatic form the essential elements of the booster mechanism. In the actual installation the piston rod of the power cylinder is attached to the frame of the vehicle, the cylinder body is jointed to the drag link of the front axle steering gear, and the signal shaft is jointed to the pitman arm actuated by the steering column. The drawing shows two positions of the signal shaft corresponding to displacements of the steering wheel, and illustrates the flow of oil through the control valve and the corresponding application of oil pressures to the operating cylinder in the directions to cause the cylinders and front wheels to follow up the motion of the signal shaft. Oil pressure is supplied to the booster through the line marked P_H leading from a constant pressure source—usually an engine-driven pump in series with some means of maintaining constant pressure such as a relief valve. The line marked P_L returns the oil to the pump reservoir.

The following typical proportions for the servo are assumed and calculations made of the time of responce to a suddenly applied displacement of the shaft:

Area (smaller of two sides) of servo piston $= a_m = 5$ square inches; difference in effective areas (a_m and a_m') of the two sides of the piston will be neglected, since it unnecessarily complicates the equations of continuity; and the error, resulting from the assumption that both areas are equal (if the piston rod is small compared to the piston) in the calculation of the transient, is negligible.

Area of the supply and return lines and passages $= a_1 = .292$ square inches

Total length of supply and return lines $.. = L = 400$ pipe diameters $= 400d$

TABLE I

Integration of Equation 2

t	x_e	x_i	Δx	x_o	v_m	$\frac{\Delta x}{v_{avg.}}$	t	t	C	P_1	P_2	F_e
	(in.)	(in.)	(in.)	(in.)	(in.)	(in./sec.)	(secs.)	(secs.)		(psi.)	(psi.)	(lbs.)
0	.2500	0		.2500	0			0		800	0	3500
			.0625	.2188	5.10	.0123	.0123		.312	736	4	3460
1	.1875	.0625		.1875	10.20			.0123				
			.0470	.1640	11.75	.0040	.0040		.382	773	27	3230
2	.1405	.1095		.1405	13.30			.0163				
			.0350	.1230	14.20	.0025	.0025		.510	748	52	2980
3	.1055	.1445		.1055	15.10			.0188				
			.0265	.0923	15.60	.0017	.0017		.729	711	89	2610
4	.0790	.1710		.0790	16.10			.0205				
			.0200	.0690	16.35	.0012	.0012		1.129	648	152	1980
5	.0590	.1910		.0590	16.60			.0217				
			.0150	.0515	16.71	.0009	.0009		1.844	542	258	920
6	.0440	.2060		.0440	16.81			.0226	2.454	453	347	30
			.0110	.0385	16.76	.00066	.00063		3.131	361	439	—890
7	.0330	.2170		.0330	16.70			.0232				
8a	.0275	.2225	.0110	.0275	16.35	.00067	.00066	.0236	5.951	4	796	—4480
8	.0220	.2280		.0220	16.00							

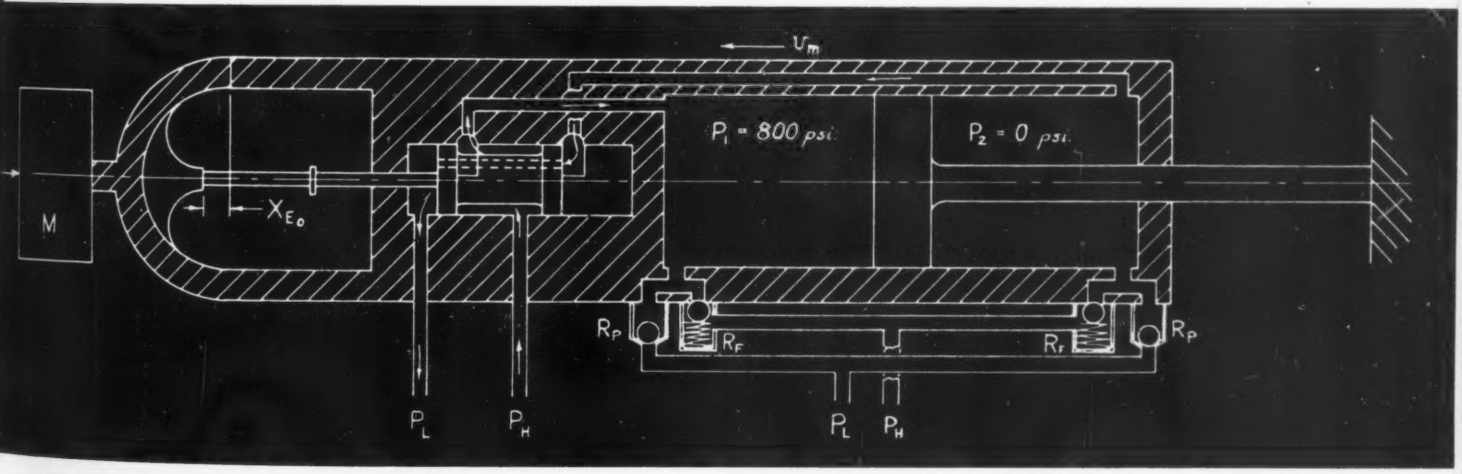
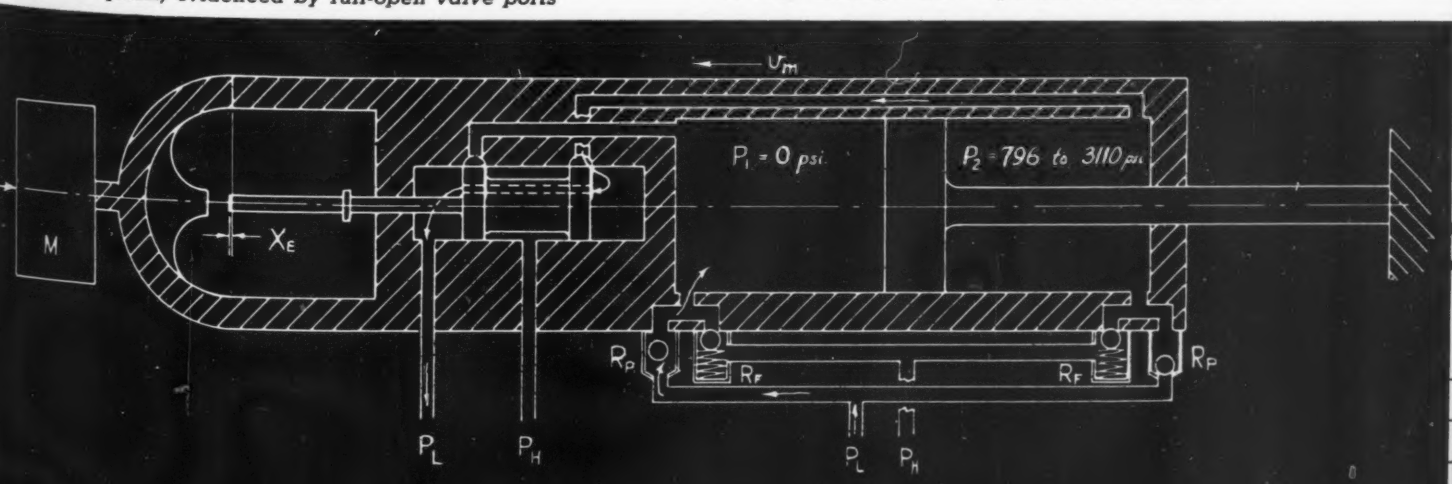


Fig. 2—Above—One-quarter-inch displacement of the signal shaft results in instantaneous response of the system, evidenced by full-open valve ports

Fig. 3—Below—Opening of replenishing valve serves to keep driving side of cylinder filled with oil



Resisting force of front wheels on drag link = 500 pounds

Inertia of front wheel and attached parts referred to the end of the drag link = $1500/g$ slugs

Number of contractions and enlargements in the complete circuit from the source of supply pressure to reservior = $n = 10$

Energy loss coefficient (average) for contraction or enlargement in circuit = $K = .75$

Constant supply pressure = $P_H = 800$ pounds per square inch (gage)

Return pressure (atmospheric) = $P_L = 0$ pounds per square inch (gage)

Friction factor for pipe flow = $f = .04$

Density of oil = $\rho = (62.4 \times .9)/32.2$ slugs per cubic foot

Coefficient of discharge of the valve orifices = $C_d = .6$

Full open area of each valve orifice = $a_o(\max) = a_l$

Length of each full open valve orifice = $x_o(\max) = .25$-inch

Length of valve lands = .25-inch

As an index to the performance of this particular design in responding to signal displacements, it will be observed how it reacts to an instantane-

TABLE II

Integration of Equation 9

i	x_e (in.)	x_i (in.)	Δx (in.)	x_o (in.)	v_m (in./sec.)	$\frac{x}{V_{avg}}$ (secs.)	Δt (secs.)	t (secs.)	C	P_1 (psi.)	P_2 (psi.)	F_e (lbs.)
8a	.0275	.2225	.0110	.0275	16.35	.00067	.00066	.0236	5.951	4	796	—4480
			.0050	.0250	16.20	.00031	.00032		7.121	0	1870	—9850
9	.0225	.2275		.0225	15.95			.0239				
			.0030	.0210	15.60	.00019	.00019		10.026	0	2440	—12700
10	.0195	.2305		.0195	15.25			.0241	11.651	0	2710	—14050
			.0020	.0185	15.00	.00013	.00013		12.831	0	2890	—14950
11	.0175	.2325		.0175	14.75			.0242	14.301	0	3110	—16050

TABLE III

Calculations in Vicinity of Dead Center

i	x_e (in.)	x_i (in.)	Δx (in.)	x_o (in.)	v_m (in./sec.)	$\frac{x}{v_{ave}}$ (secs.)	Δt (secs.)	t (secs.)	C	P_1 (psi.)	P_2 (psi.)	F_e (lbs.)
11	.0175	.2325		.0175	14.75			.0242	14.30	0	3110	16,050
			.0175				.00148			0	3200	16,500
12	.0000	.2500			8.88			.0257		0	3200	16,500
				.0050			.00066					
13	.0050	.2550			6.28			.0264	86.50	0	3200	16,500
				.0010			.00017					
14	.0060	.2560		.0060	5.62			.0266	60.10	0	2700	14,000

ously applied change in position of the signal shaft equal to .25-inch, the signal being held stationary after the .25-inch displacement. This is illustrated in *Fig.* 2, showing the valve ports in the full-open position at the instant following displacement of the signal. Full pressure is immediately transmitted to the left chamber of the work cylinder, which begins to move in the direction to catch up with the signal. The valve ports will of course begin to close (and commence throttling of the oil) as the cylinder changes position in response to servo signal—in the same

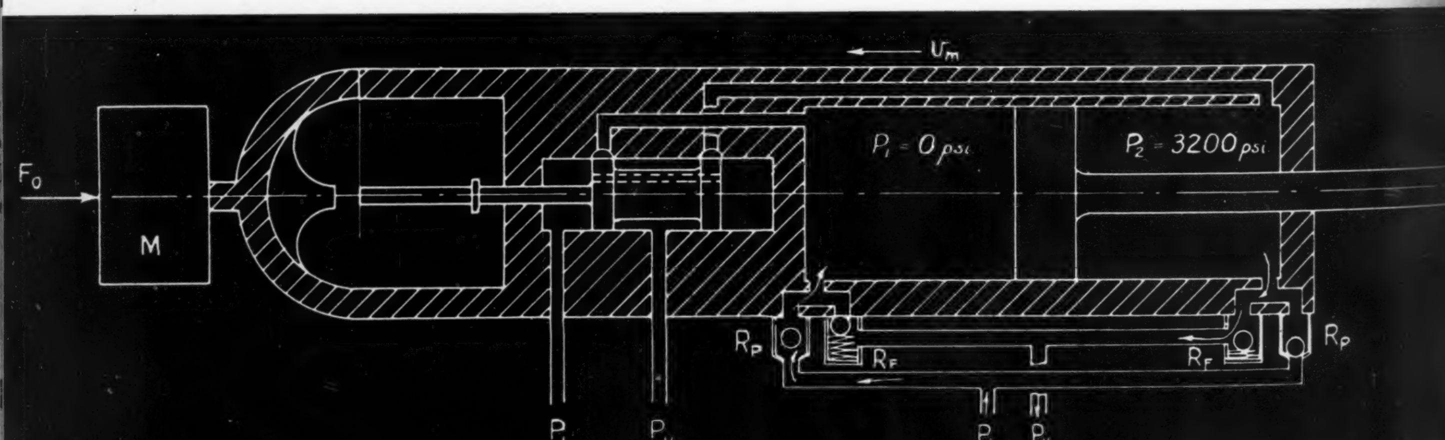
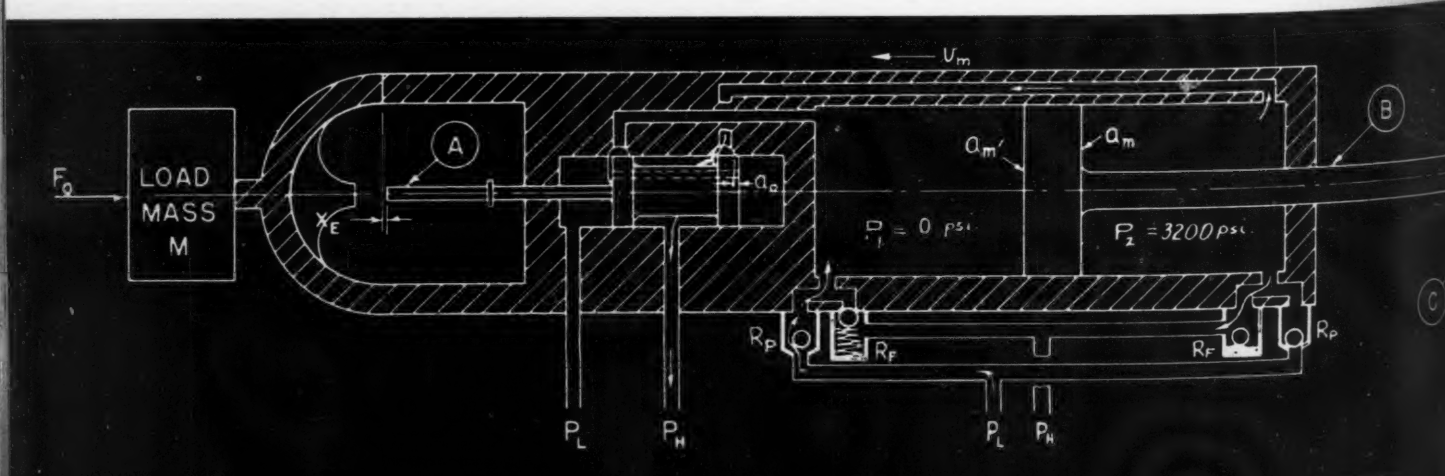


Fig. 4—Above—When the back pressure of the return circuit exceeds 3200 pounds per square inch the relief valve opens, maintaining this pressure

Fig. 5—Below—In combination with Fig. 4 are illustrated the limiting positions for normal operation of the relief valve, protecting against excessive pressures



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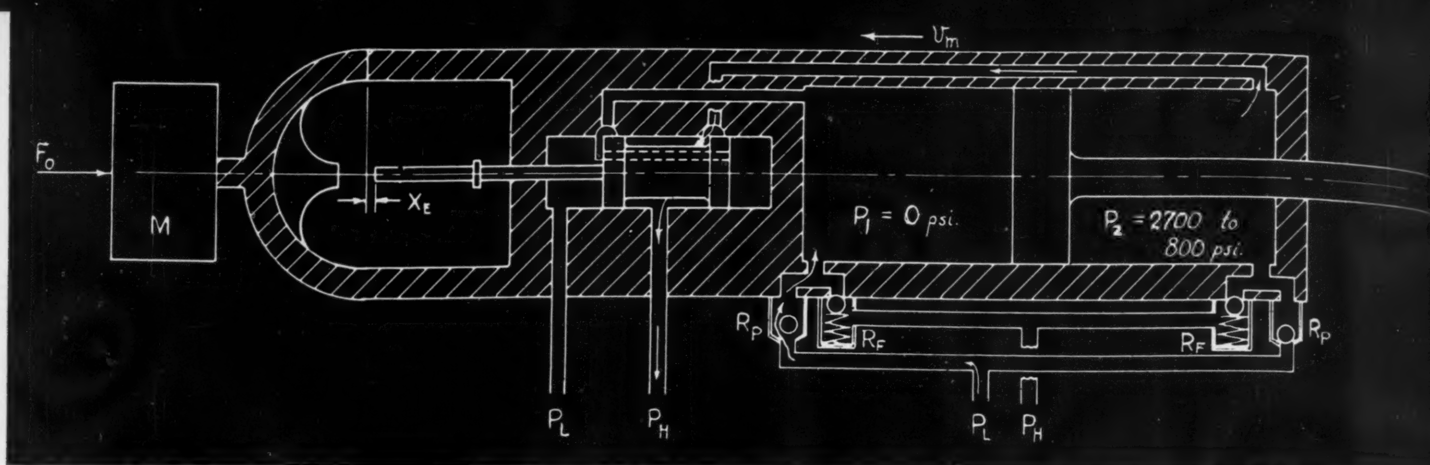

Fig. 6—When the driving pressure falls below 3200 pounds per square inch, deceleration is not constant

amount as the movement of cylinder.

As derived in the previous article, the differential equation which describes the motion of the servo-motor (assuming a leakless piston and valve) is

$$\frac{d(v_m)}{\left(P_H-P_l-\frac{F_0}{a_m}\right)-\frac{\rho}{2}\frac{a_m^2}{a_l^2}\left[f\frac{L}{d}+nK+\frac{2}{C_d^2}\left(\frac{a_l^2}{a_o^2}-1\right)\right]v_m^2}=\frac{a_m}{M}dt \quad \ldots\ldots\ldots(1)$$

where v_m is the velocity of the servo-motor cylinder.

Substituting the values of the equipment proportions and the other constants defined above, this becomes (values are converted to inch units)

$$\frac{d(v_m)}{700-Cv_m^2}=\frac{d(v_m)}{700-\left[.291+.069\left(\frac{a_l^2}{a_o^2}-1\right)\right]v_m^2}=\frac{5}{4.17}dt \quad \ldots\ldots\ldots(2)$$

Integrating Equation 2 under the assumption that over a short interval of time the valve opening is constant (therefore C, the coefficient of v_m^2, which is a function of the valve opening at any given instant of time, is also constant) gives

$$t_{i+1}-t_i=\Delta t=-\frac{.0157}{\sqrt{C}}\left[log_e\frac{v_m\sqrt{C}-\sqrt{700}}{v_m\sqrt{C}+\sqrt{700}}\right]_{v_i}^{v_{i+1}} \quad \ldots\ldots\ldots(3)$$

where t is in seconds, and v_m in inches per second. Hence

$$C=.291+.069\left[\left(\frac{.25}{x_0}\right)^2-1\right]$$

in which $.25/x_o$ has been substituted for a_l/a_o, the ratio of full area of valve orifice opening to the orifice area at partial opening. The ratio is the same in both cases if the ports in the valve housing are rectangular in shape, which may, for convenience, be assumed in this case. The quantity Cv_m^2 in Equation 2 will be recognized as the total pressure drop due to pipe loss in the complete pipe and passage circuit, plus the throttling drop due to the two valve orifices. $(C/2)v_m^2$ is then equal to the drop on account of one-half the circuit, i.e., from the pressure supply into the cylinder chamber, or equal to the drop from the other side of the servo-cylinder back to the reservoir of the pressure supply.

At this point attention is called to TABLE I which is a tabulation of the data obtained in the step-by-step integration of Equation 2 over successive small intervals of time through the use of Equation 3.

Headings used are defined as follows:

i = Beginning of any given time interval; (then $i+1$ refers to the beginning of the next succeeding time interval)

x_e = difference in positions of signal and servo-cylinder (called error)

x_i = distance traveled by the servo-cylinder from the time corrective motion begins

Δx = distance moved by the servo-cylinder during a given small interval of time = $x_{i+1}-x_i$

x_o = length of valve opening

v_i = velocity at the beginning of a given time interval (v_{i+1} is the velocity at the end of the same time interval)

v_{avg} = average velocity over a given time interval—assumed to be the arithmetic mean of the velocities at beginning and end

$\Delta x/v_{avg}$ = small time interval calculated necessary to traverse a given increment of displacement Δx

Δt = small time interval calculated by Equation 3 necessary for the velocity change $v_{i+1}-v_i$ to take place under the condition of the differential Equation 2.

Repeated estimates must be made of

TABLE IV

Integration of Equation 11

i	x_e (in.)	x_i (in.)	Δx (in.)	x_o (in.)	v_m (in./sec.)	$\frac{\Delta x}{v_{ave}}$ (secs.)	Δt (secs.)	t (secs.)	C	P_1 (psi.)	P_2 (psi.)	F_e (lbs.)
14	.0060	.2560		.0060	5.62			.0266	60.10	0	2700	14,000
			.0020	.0070	5.14	.0C039	.00039		44.30	0	1970	10,350
15	.0080	.2580		.0080	4.66			.0270				
			.0020	.0090	4.28	.00047	.00047		26.77	0	1290	6,950
16	.0100	.2600		.0100	3.89			.0275				
			.0020	.0110	3.51	.00057	.00057		17.95	0	976	5,350
17	.0120	.2620		.0120	3.13			.0281				
			.0020	.0130	2.69	.00074	.00074		12.89		890	4,950
18	.0140	.2640		.0140	2.25			.0288				
			.0020	.0150	1.58	.00127	.00125		9.69		824	4,720
19	.0160	.2660		.0160	0.90			.0301				


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v_{i+1} at the end of an interval. When the estimated end velocity v_{i+1}, in combination with the known velocity at the beginning of the interval v_i yields the same time interval in the form Δ_x/v_{avg}, as when obtained by substitution in Equation 3, the velocity v_{i+1} has been chosen correctly and the corresponding quantities may be entered in TABLE I.

To illustrate the method of carrying through the integration over small time intervals, let the first interval 0 to 1 be considered. At $i=0$ (the beginning of the interval) the error $x_e=.2500$-inch; distance traveled $x_i=0$; valve opening $x_o=x_{o(max)}=.2500$-inch; velocity of the servo-cylinder $v_m=0$; no time has elapsed, therefore $t=0$. Driving pressure $P_1=P_H=800$ pounds per square inch; retarding pressure $P_2=P_L=0$; and the net accelerating force

$$F_e=(P_1-P_2)a_m-F_0=800\times5-500=3500 \text{ pounds} \quad (5)$$

During the first time interval $i=0$ to $i=1$, it will be prescribed that the servo-cylinder shall recover $\Delta_x=.0625$-inch, which is then entered in TABLE I between $i=0$ and $i=1$. This then fixes for the end of the interval (at $i=1$): $x_e=$error $=.1875$-inch, $x_i=.0625$, and valve opening $x_o=.1875$-inch. The servo-cylinder velocity v_{m_1} at $i=1$ must be determined by trial and error. Estimating that it will be 10.2 inches per second, the average velocity over the interval would then be 5.1 inches per second, and the time consumed during the interval would be $\Delta x/v_{avg}=.0625/5.1=.0123$ seconds. This is temporarily recorded. To check the assumption of the end velocity, Δt is calculated by substituting in Equation 3 which gives the time necessary to change from the beginning velocity $v_{m_0}=0$ to $v_{m_1}=10.2$, under the accelerating force.

Evaluated for Mean Value of Valve

In substituting in Equation 3 it is necessary to evaluate C, which is considered constant over the time interval, i.e., it is evaluated for the mean value of the valve opening during the interval $i=0$ to $i=1$, $x_o=.2188$-inch.

Evaluating C for the interval, gives

$$C_{0-1}=.291+.069\left[\left(\frac{.2500}{.2188}\right)^2-1\right]=.312 \quad (6)$$

Substituting in Equation 3 gives

$$\Delta t=-\frac{.0157}{\sqrt{.312}}\log_e\left[\left(\frac{10.2\sqrt{.312}-\sqrt{700}}{10.2\sqrt{.312}+\sqrt{700}}\right)\times\left(\frac{0\sqrt{.312}+\sqrt{700}}{0\sqrt{.312}-\sqrt{700}}\right)\right]=.0123\, seconds \quad (7)$$

which agrees with the previous time determined necessary to traverse the distance. The assumed velocity v_{m_1} was therefore correct and the rest of the conditions at $i=1$ may be determined and entered into TABLE I. For instance, $P_1=800-(C/2)v_{m_1}{}^2=796$ pounds per square inch; and $P_2=(C/2)v_{m_1}{}^2=4$ pounds per square inch. F_e may be calculated from Equation 5.

Some discretion must be exercised in specifying the distances Δx for which the time intervals are calculated. Generally speaking they should be roughly proportional to the amount of valve opening existing at the beginning of the interval. The accuracy of the step integration is not affected much by large steps taken when the orifice is large, but as it becomes small (valve nearly closed) the sensitivity to approximation increases rapidly.

Repeat for Smaller Valve Movements

The trial and error method of establishing succeeding velocities is then repeated for successively smaller movements of the valve until the valve is closed off to the point where P_1 is observed to reach 0, i.e., when $(C/2)v_m{}^2=800$ pounds per square inch, which occurs at nearly station $i=8a$ in TABLE I.

With further closure of the valve the value of $C/2$ becomes larger and tends to reduce P_1 even lower, i.e., below atmospheric, and this will occur so long as $(C/2)v_m{}^2>800$ pounds per square inch.

In order to keep the driving side of the cylinder filled with oil and to maintain a known pressure there, a replenishing valve R_p will now be assumed to open, as in *Fig.* 3, which will maintain $P_1=0$ so long as $(C/2)v_m{}^2>800$ before dead center is reached. In the case of overshoot (travel beyond dead center of the valve) the replenishing valve can be counted on to keep $P_1=0$ so long as $(C/2)v_m{}^2>0$, or, in other words, until the servo-cylinder comes to momentary rest at the maximum distance of overshoot.

So long as P_1 remains constant at 0, Equation 2 must be entered with different values than those used for TABLE I calculations, for the new effective pressures involved. *Fig.* 3 shows the conditions of flow.

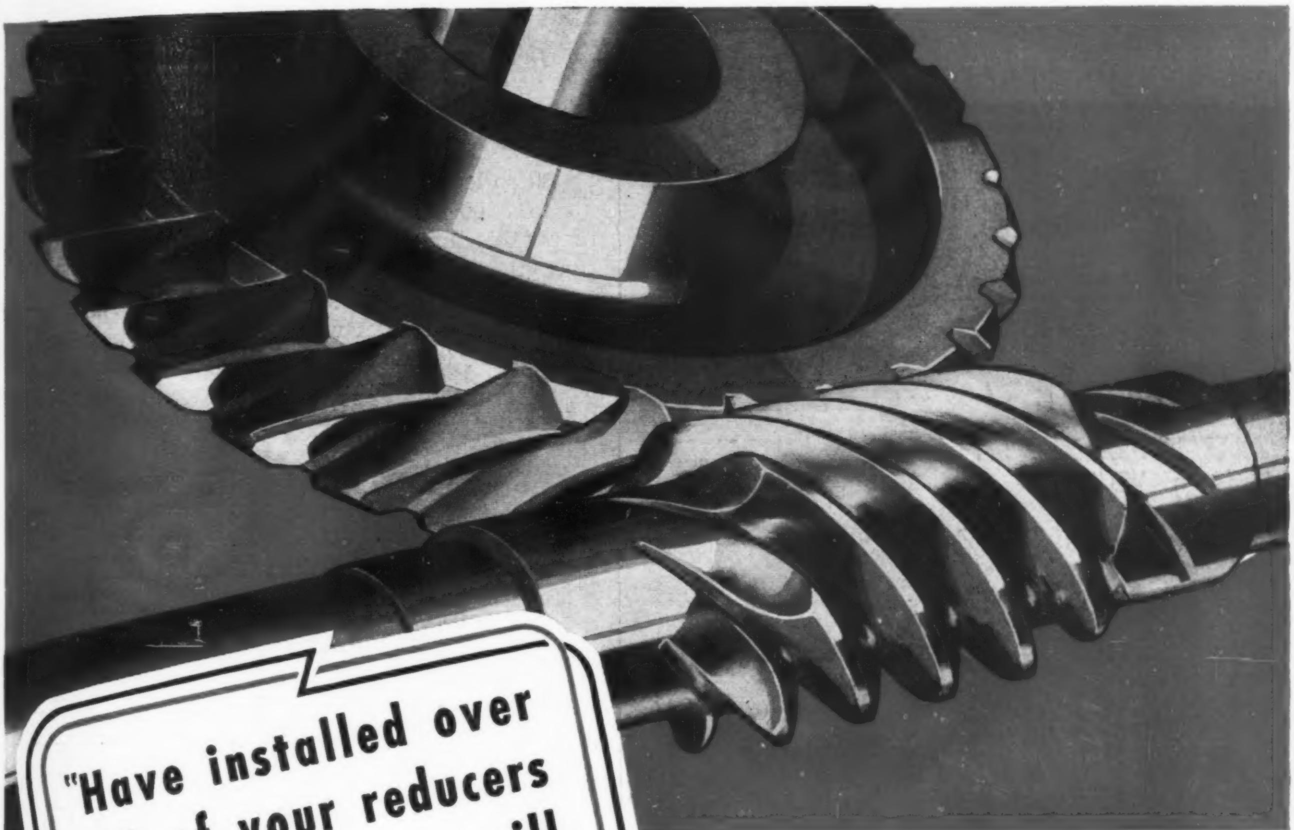
Drawing of Curves Recommended

In order to anticipate the variations in pressure and velocity, it is recommended that curves be plotted of P_1, P_2, v_m, and x_e against time t as the calculations progress. They are shown in *Figs.* 7 and 8 for the calculations in this case.

Substituting in Equation 2 $P_H=0$ (in effect, it is cancelled out) since $P_1=0$; and further substituting $(C/2)v_m{}^2$ in place of $Cv_m{}^2$ since the variable pressure drop in the circuit is now confined to the return side, there results

$$\frac{d(v_m)}{-100\frac{C}{2}v_m{}^2}=\frac{a_m}{M}dt \quad (8)$$

the solution of which is



$$\Delta t = -\frac{4.17}{5\sqrt{100\frac{C}{2}}}\left[\tan^{-1}\sqrt{\frac{C/2}{100}}v_m\right]_{v_i}^{v_{i+1}} \qquad (9)$$

TABLE II represents the integration using Equation 9 so long as P_1 remains 0, and until P_2 builds up to 3200 pounds per square inch which is a reasonable limit to which to hold the cylinder for strength.

At this point (when P_2 reaches 3200 pounds per square inch) a relief valve R_f is presumed to blow and hold P_2 to 3200 pounds per square inch so long as the back pressure of the return circuit$>$3200 pounds per square inch. *Figs.* 4 and 5 show such

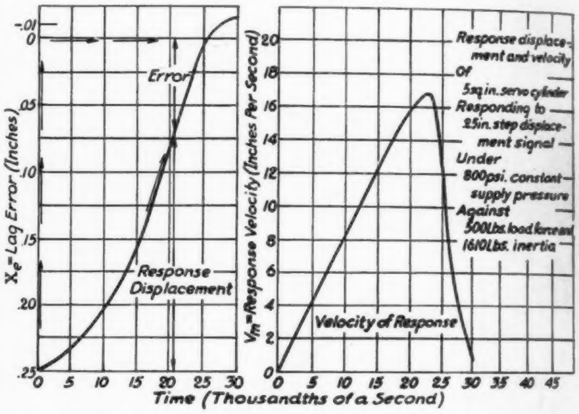


Fig. 7—Response of the circuit to an instantaneous signal is practically complete in 1/40-second

conditions which in this case occur from a point just before dead center until some distance beyond dead center. The calculations during this period are represented in TABLE III and simply consist of constant deceleration under the constant effective force F_e=16,500 pounds.

A constant check on P_2 as governed by the resistance of the return circuit, should be made during the calculations of TABLE III. When P_2, as calculated by the expression, (beyond dead center $=800+(C/2)v_m^2$ falls to less than 3200, the deceleration is no longer constant.

From this point on the differential equation is

$$\frac{d(v_m)}{-800-100-\frac{C}{2}v_m^2} = \frac{a_m}{M}dt \qquad (10)$$

The solution of Equation 10 is

$$\Delta t = -\frac{4.17}{5\sqrt{900\frac{C}{2}}}\left[\tan^{-1}\sqrt{\frac{C/2}{900}}v_m\right]_{v_i}^{v_{i+1}} \qquad (11)$$

which is applied until the cylinder comes practically to rest. TABLE IV shows the results. The conditions are shown in *Fig.* 6.

Inspection of the transient shown in *Fig.* 7 indicates that response to the severe test of an instantaneous signal is extremely rapid. Practically complete correction is made in 1/40 seconds and over-

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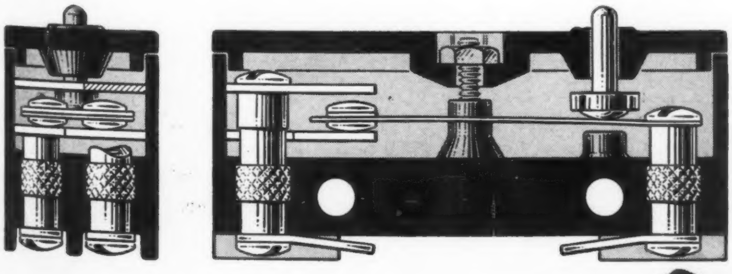

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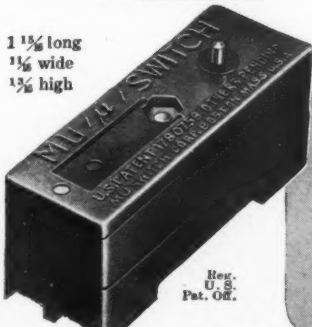

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shoot and oscillation damped out. This design appears to be quite satisfactory for the purpose although the response to other types of signals, such as a uniform velocity, might be checked also to obtain a more complete picture.

In actual practice the relief and replenishing valves are not always used. Valve lands are often shaped and relieved to avoid the sharp cutoff observed in this case, with the consequent high pressure rise. However, damping with a relieved valve land is liable to be considerably less effective and permit more overshoot and oscillation than observed in this case.

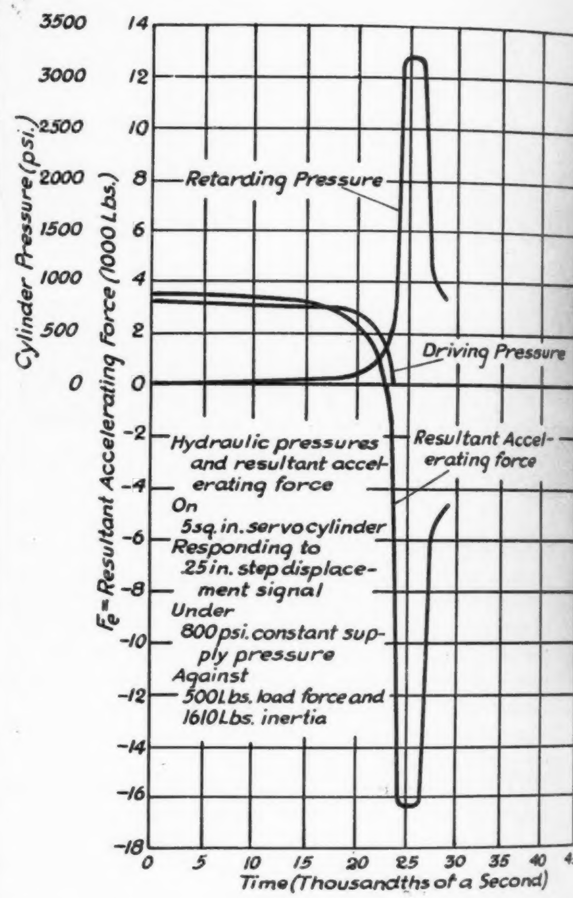

Fig. 8—Supplementing the velocity curve in Fig. 7, this curve illustrates force and pressure conditions as a function of time and resultant accelerating force

A practical question is often raised as to the reliability of such a servo in a steering mechanism if the hydraulic pressure should fail. In case of pressure loss the signal shaft, attached to the pitman arm, bottoms against the cylinder housing beyond a certain small error and then the control is entirely mechanical.

Amount of permissible error, or backlash, in an actual steering booster is preferably somewhat smaller than the + ¼-inch indicated in the example of the response characteristic. The maximum valve opening may be reduced by making ports shorter and wider, but maintaining a total area equal or slightly greater than the area of the supply passages.

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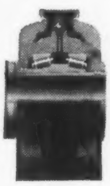

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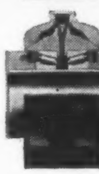
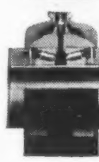


• Dodge-Timken Clamp Sleeve Bearings . . . Hanger Bearings . . . Pillow Blocks and Unit Mounts. Rugged, and dependable general purpose bearings.



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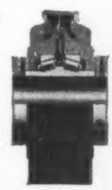
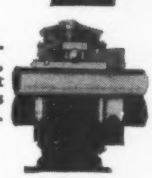
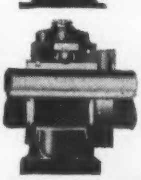


• Dodge-Timken "Double Interlock" Pillow Blocks and Unit Mounts for normal radial, thrust and shock loads.



• Dodge "DH" Ball Bearings — Hanger Bearings, Pillow Blocks and Unit Mounts. For high speeds and light to moderate loads.



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Pneumatics in Typewriters

(*Continued from Page* 102)

a round valve on an impulse canceling device to the select reverse valve in the valve box. This permits canceling two impulses, since the record moves forward to the next series of selection perforations and then reverses, and synchronizes the reading of a dial indicator, mounted next to the pushbuttons, with the actual position of the record. To each of these pneumatics is connected an electrical contact that is closed only with the final stroke of the pneumatic, thereby completing the circuit for the selector magnet after the movable contact has arrived on a stationary contact that is wired to the corresponding pushbutton depressed for selection.

Electric contact consists of five rows of pushbuttons, each row having a white button for clearing that row in case one or more buttons are pushed in error, or a new selection is to be made. The pushbuttons, when depressed, close the circuit for the selector magnet. Each pushbutton contact is connected to a stationary conductor on the contact stepper plate.

Directional Switch Is Set Manually

A record directional switch, mounted outside the dial indicator before the operator, determines which direction the record should progress over the tracker bar—forward, reverse or repeat. It is set manually in the desired direction, to the extreme right for forward, center for repeat and to the extreme left for reverse. Shifting is accomplished by three pneumatics. Valves for these pneumatics, each one of which is opened by turning the switch to one of the above positions, receive the impulse through the select perforations passing over the tracker bar. They complete that circuit through a nipple slide or cut-off valve above the selector magnet. When this nipple valve is operated, it interrupts the select impulse, and at the same time establishes an impulse with the repeat valve in the valve box. This cut-off valve is the only deviation from the original select impulse circuit. When the record directional switch is in center position, a second impulse channel provides a connection between the tracker bar rewind slot and the select reverse valve in the valve box.

Reverse selection cut-off switch is located on the bottom of each panel. It is an ordinary electric switch to cut off the circuit to the selector magnet when the record is running in reverse direction, making it possible to reverse to the first selection without clearing the pushbuttons.

Reverse selection canceling and selector magnet unit is mounted on the bottom of the desk, toward the front. It has two distinct functions: First, to cancel two select impulses in reverse to maintain synchronization of the dial indicator and position of the record on the tracker bar; second, to

(*Concluded on Page* 140)

(*Concluded from Page* 134)

establish an air impulse to the repeat or transmission reset pneumatic valve at the time the movable stepper contact meets the stationary contact corresponding to the button pushed. The second function is independent and has no connection with the canceling function, but the unit is so designed for convenience in construction.

Canceling unit consists of a round stepper valve which interrupts the suction from the reverse select valve in the valve box to the reverse stepper pneumatic for two steps. The first two suction impulses are utilized to step this round valve by means of a stepper pneumatic to a position whereby a suction connection with the reverse select stepper pneumatic is established when the third group of select perforations are passing over the tracker bar. This procedure synchronizes the paragraph readings as marked on the record with the dial indicator position before the pointer steps backwards. The round valve is kept in this advanced position by a detent lever until the record has arrived at the selected paragraph. This valve is brought to its neutral position under the influence of a coil spring when the lever is freed.

Nipple Valve Interrupts Impulse

The selector magnet unit, located on the same frame with the canceling unit, consists of a nipple valve, electric magnet and a nipple valve reset pneumatic. The select impulse is connected through the nipple valve to the directional switch valve, and is actuated by the electric magnet whenever the record arrives at a select position. Its electrical impulse is received whenever the circuit is closed by a depressed contact on the pushbutton bank. When the nipple valve is operated by the magnet, it interrupts the select impulse. At the same time, it establishes an air impulse to operate the "repeat" or transmission reset valve, and provides suction to a pneumatic that shifts the transmission into neutral position, permitting the record to run forward to the stop at the selected paragraph. Immediately preceding the stop perforation in the record is a perforation which, through a pneumatic, resets the nipple valve and the magnet.

An emergency stop unit or pneumatic relay, located on the bottom of the case, consists of a simple primary valve with two electrical contacts. It is operated through the repeat impulse from the tracker bar. Operation of this primary valve closes the electrical circuit to the magnet, moving the nipple valve, and thereby shifting the transmission into a neutral position.

Suction cut-off block for transmission forward and reverse selector pneumatics is located directly under the transmission frame. Its function is to cut off the suction to either pneumatic.

Pneumatic actuation of a complicated mechanical device has proved practical in this machine, indicating pneumatics are worthy of investigation in connection with design problems of other mechanisms. This power method particularly adapts itself to quiet operation so often desired.

Playing Hob with Involute Gears

(*Continued from Page* 109)

tangency of the line of action. There can be no gear action beyond this point and the result is that the end of the tooth cuts away a small portion of the involute profile. While actually little of the profile is removed, a disproportionately large amount of the line of action is cut off. The following formulas from Buckingham's *Manual of Gear Design* give the depth to which a gear may be hobbed without introducing undercut and a close approximation to the amount of involute profile removed when the gear is cut to a greater depth.

$$R = \frac{N}{2P} \qquad (56)$$

$$R_u = R \cos^2 \phi - y(1 - \sin \phi) \qquad (57)$$

$$x = \frac{(R_u - R_r)^2}{6R \cos \phi \sin^2 \phi} \qquad (63)$$

Considering the elimination of undercut it should be clear that in a pair of gears the center distance should be equal to, or greater than, the sum of the two undercut radii plus the tooth depth, plus the clearance, if the gears are cut on nonstandard diameters. Using the aforementioned formula for undercut radius it is seen that with standard tooth depths, if the total number of teeth in the pair of gears is greater than forty for a 20-degree pressure angle or than seventy-five for a 14½-degree pressure angle, the gears can be cut to run at standard center distance and pressure angle without undercut. This may, of course, result in a decreased outside diameter with a larger gear and an increased outside diameter with a smaller gear.

Results in Stronger Tooth Form

The outside diameters may be figured by adding twice the tooth depth to the undercut diameter of the smaller gear and reducing the outside diameter of the larger gear from the standard outside diameter by the same amount that the smaller gear has been increased over standard. The phantom rack shown in *Fig.* 5 would then mate with these two gears on the same line of action, and the pressure angle would remain unchanged. Where the total number of teeth in the pair of gears is less than the value mentioned it will be necessary, if undercut is to be eliminated, to cut one or both on oversize diameters and to increase the center distance and pressure angle.

To return to *Fig.* 5, since the gear on the right has not been cut below its undercut radius, though cut in to the same depth from its outside diameter by the same hob, it can be seen to have useful working surface for its full profile without undercut, as well as being a far stronger form. Obviously each of these gears will run correctly with the rack, and since the two gears have the same

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number of teeth they will have the same pitch diameters and pressure angles. Consequently they will have the same base circle. They will, of course, have the same base pitch as the rack.

It is reasonable to ask then how they will run with each other. From the crossed belt analogy it is evident that they should run quite satisfactorily, though at increased pressure angle. However, at the center distances shown in *Fig.* 5 there is a space between the adjacent faces of the two teeth due to the fact that they do not have the same lines of action with the rack. There would also be an identical space between the other faces of the teeth and the adjacent faces of the next teeth which are not shown. With the two gears running together the driving faces would, of course, be in contact, taking up this space and increasing the space between the nondriving faces by the same amount. In cutting the right-hand gear its center distance has been increased by the same amount that the outside diameter has been increased, leaving the depth of cut standard. Therefore, if these two gears run on this increased center distance a certain amount of backlash will have been produced, though far less than if they had both been standard gears and run on these centers. This backlash is given by the following formula:

$$b = 2c_r \cos\phi_r\ (inv\ \phi_r - inv\ \phi) - \frac{n_1+n_2}{P}\sin\phi$$

$$= 2(R_{b1}+R_{b2})\ \ (inv\phi_r - inv\ \phi) - \frac{n_1+n_2}{P}\sin\phi$$

In considering backlash, it should be remembered that a certain amount is necessary for good gear action. In the first place the film of lubricant between the tooth faces has thickness. In the second place, no gear can be absolutely perfect and sufficient backlash must be allowed so that the gears will not mesh too tightly because of eccentricity or errors in tooth profile. This backlash is produced in standard gears by sinking the hob deeper into the blank in one or both of the gears, the formula (from Buckingham's *Manual of Gear Design*) for this being, when both gears are cut deeper

$$i = \frac{b}{4\ \sin\phi} = \begin{cases} b \text{ for } \phi = 14\tfrac{1}{2} \text{ degrees} \\ .73b \text{ for } \phi = 20 \text{ degrees} \end{cases}$$

and when only one gear is cut deeper

$$i = \frac{b}{2\sin\phi} = \begin{cases} 2b \text{ for } \phi = 14\tfrac{1}{2} \text{ degrees} \\ 1.46b \text{ for } \phi = 20 \text{ degrees} \end{cases}$$

As to the amount of backlash desirable, A. C. Rasmussen in the February, 1939, issue of *Product Engineering* gives for pitch line velocities less than 1,000 feet per minute, the value $b = .030/P$ and for greater pitch line values, $b = [.030 + 3\times10^{-6}(v - 1000)]/P$. The A.G.M.A. also gives a table of backlash, Table I, for general use in gearing running not over 1500 feet per minute with which these formulas agree reasonably well.

If the increase in center distance does not exceed $1/2P$ the backlash produced will not be excessive, in general, with special gears of the 20-


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degree system, which makes the cutting of small pinions one tooth oversize a simple and convenient method of eliminating undercut and strengthening the tooth. In fact, where the drive is always from one direction or where speeds are not high, it is difficult to say when, if ever, backlash would be considered excessive. On the other hand, where the driven gear and apparatus connected to it are likely to overrun the driver, reversing the load on the teeth, it is desirable to keep the backlash down to a minimum, as for instance in a motor drive where a brake on the motor is used to stop rotation when power is shut off. In this case it is usually desirable to use a nonmetallic pinion or a shock-absorbing coupling. Another case where it is at least desirable to know the effect of backlash is that in which gears are used to control the motion or position of one part in respect to another, as in a camshaft drive. Here it is desirable to know the possible angular error due to backlash. The term angular backlash will be used to describe the angle through which one gear of a pair can be turned while the other pair is held fixed and is $\beta = b/R_b$ radians.

Where it is desired to eliminate the backlash created by these special gears they may be run on a center distance that is less than the sum of the cutting center distances. The outside diameters of both gears, however, must be reduced by the amount the center distance has been reduced to maintain the clearance between the tip of the tooth of one gear and the root of the tooth space of the other. The formulas for this case are as follows:

TABLE I

A.G.M.A. Recommended Backlash Limits

Pitch	Backlash min.	Backlash aver.	Backlash max.
1	.030	.040	.050
1½	.020	.027	.033
2	.015	.020	.025
2½	.012	.016	.020
3	.010	.013	.017
4	.008	.010	.013
5	.006	.008	.010
6	.005	.007	.008
8	.004	.005	.006
10	.003	.004	.005
12	.003	.004	.005
16	.002	.003	.004
24 and finer	.002	.003	.004

$$inv\,\phi_r=\left(\frac{n_1+n_2+N_1+N_2}{N_1+N_2}\right)\tan\phi-arc\,\phi$$

$$c_r=\left(\frac{N_1+N_2}{2P}\right)\frac{\cos\phi}{\cos\phi_r}=\frac{R_{b1}+R_{b2}}{\cos\phi_r}$$

$$h=c_r-(R_{r1}+R_{r2}+clearance)$$

$$R_{o1}=c_r-\frac{N_2+n_2}{2P}$$

$$R_{o2}=c_r-\frac{N_1+n_1}{2P}$$

where $n/2P$ = increase in R_r over the standard.

Buckingham gives a table based on the 14½-de-

gree full depth rack in which gear combinations are worked out for gears of twelve to one hundred teeth. This system eliminates all undercut and maintains contact ratios of better than 1 for any pair of gears and better than 1.4 for the great majority of combinations. It has the slight disadvantage, however, that the center distances with the smaller gears are not a whole number divided by twice the diametral pitch and also that the center distance is not constant for a given total number of teeth in a pair if one gear of the pair has over forty teeth, and the other has less than forty. These two points may prove inconvenient in trains of gears, in jackshaft arrangements where the input and output shafts are in line, and in planetary gearing.

So far only hobbed or rack-generated gears have been considered. There is another method of generation exemplified by the regular Fellows machine and by the Farrell-Sykes. This type of machine uses a reciprocating cutter that has the shape of a pinion of the system to be cut. As this pinion-shape cutter is reciprocated, it is fed into depth on a blank of suitable diameter. The blank and cutter are then rotated in opposite directions and the planing action of the cutter generates correctly formed involute teeth on the blank as shown in *Fig.* 6.

Gears for Restricted Centers

Referring back to the crossed belt analogy we can see that special gears can be cut as readily with this method as with hobbing. In this case, however, the excess backlash when produced depends not only on the center distances at which the finished gears are run but also the center distances at which they are cut and the number of teeth in the cutter. In general this inherent backlash is less than with hobbing, but its calculation is considerably more complex and, as the number of teeth in various cutters varies considerably, it is better to leave the calculations to the company cutting the gears.

As has been indicated, one of the chief reasons for the use of special gears is to eliminate undercut and also to achieve a stronger tooth form. Another place where they may be desirable is the case in which the position of two shafts is fixed by other factors in the design, and standard gears cannot be worked out in standard pitches to connect them without introducing excessive backlash and decreasing the engagement of the teeth. In this case the best combination of standard gears, the sum of whose pitch radii will be less than the fixed center distances, should be determined. Then adding to the outside radius of one of these gears the difference between the sum of the pitch radii and the running center distance, this oversize gear may be cut to standard tooth depth.

A similar case is often encountered in planetary gearing. An example of this is a commercial gear reducer with a 9:1 ratio. This is a simple ring and sun gear planetary with three idlers, the ring having 96 teeth and the sun pinion 12 teeth which,

to avoid undercut, are cut on a blank having a 13-tooth outside diameter. This would seem to require a 41½-tooth idler. In the high speed lighter duty models where the idlers are made of textolite these planets are cut 41 teeth on a 41½-tooth blank to take advantage of the stronger tooth form of the oversize gear. In the heavy-duty models, where the idlers are bronze or steel, they are cut 42 teeth on the 41½-tooth blank to reduce undesirable backlash.

In this connection it should be noted that the planets in this type of planetary are true idlers, that is, the number of their teeth does not affect the gear ratio of the system. When using the formulas given by George S. Hoell, (M.D., November, 1941) with these special gears, they may be transposed to use the sun and ring gear rather than the sun and planets as a basis of calculation.

Special gears, properly designed, require no extra work or expense in their manufacture. Indeed, the man at the hobbing machine may never know he is cutting a special gear. They only require some extra work at the designer's board and the results are usually amply justified.

Miniature Motors

(*Continued from Page* 119)

lems. In this case maximum power with minimum size and weight is required. The motors are, therefore, provided with special fans for cooling to increase the power without overheating. These motors operate at as high a speed as possible to increase the power output per unit of weight. Windings of the armature require careful protection for these high speeds and must be well balanced. In many cases, too, these motors are built in as a part of the tool, the motor manufacturer supplying only the parts. This always requires close cooperation between designer and customer to insure proper mounting of the motor parts to produce the desired results.

Another interesting development is a small motor-generator designed for sound projection. A universal motor operates the projector and a direct-current generator. The output of the generator is utilized to operate the power amplifier for the sound system. By this means the sound projector can be used on either alternating or direct current. Because of the necessity of close regulation on the generator output, a governor on the motor keeps the speed constant.

Noise is Kept to a Minimum

Trends in modern domestic oil burners have been to build in the motors as part of the burners. The motor is flange mounted in many cases with the fuel pump unit in the end casting opposite the flange. Then the fuel pump is essentially a part of the motor and presents an easily serviced unit which can be replaced readily. Motor is other-

wise a standard split-phase type with emphasis placed on high starting torque. Hum of this motor is kept to a minimum as noise can easily be telegraphed through the ducts or pipes of the heating system.

In the same class as oil burner motors are small motors for hot water circulators. These motors have essentially the same motor parts except for the bearing construction. Being mounted vertically, they require special oil slingers and oil reservoir construction to prevent the oil running down the shaft. They also must have ample power as operation may be nearly continuous and the duty is heavy. Cool running is essential for this application.

New Magnet Alloys Increase Use

A rather new entry into the field is the small permanent magnet type of motor. With the advent of better permanent magnet alloys these motors have been made more feasible. Their use is limited to places where the high cost of the magnet alloy can be offset by the other advantages. Since the motor is a direct-current shunt type with the field supplied by the magnet, the efficiency is considerably greater than that of a motor of similar power with conventional windings. This may be important, as when operating a small motor from dry cells. Eliminating the field coils also permits the motor to occupy much less space.

With the advent of wartime needs replacing the domestic production, the motor designer is now confronted with a different set of requirements. Emphasis is now placed on quality and reliability, involving closer limits and more exacting manufacture. Most government applications require ball bearings lubricated for —60 to 125 degrees Fahr. instead of the more familiar sleeve bearings.

Airplanes Require Special Designs

Design of these motors is even more specialized than peace-time requirements. Each motor has a particular job to do and is built to do this job alone. In most applications, such as airplane work —the specific nature of which cannot now be divulged—the last ounce of efficiency is exacted from the motor to keep the weight and power required as low as possible. This involves high speeds with extra full windings. These windings might ordinarily be considered poor commercial practice from a normal use standpoint. Ball bearings necessitate careful alignment with close machining tolerances. This automatically eliminates the stamped cases so prevalently used on commercial work, as well as the ordinary tools and machines for this type of work.

Thus, fractional horsepower motors for war work, because of their specialized requirements, cannot be compared to the mass production commercial types. In many cases the changeover requires extensive retooling and new machines to meet exacting specifications and design.

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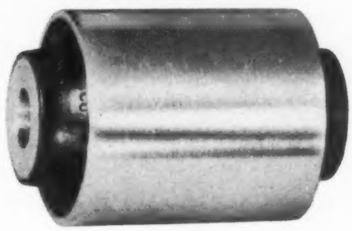
Take up { PARALLEL MISALIGNMENT / ANGULAR MISALIGNMENT / WITH LOW POWER LOSS }

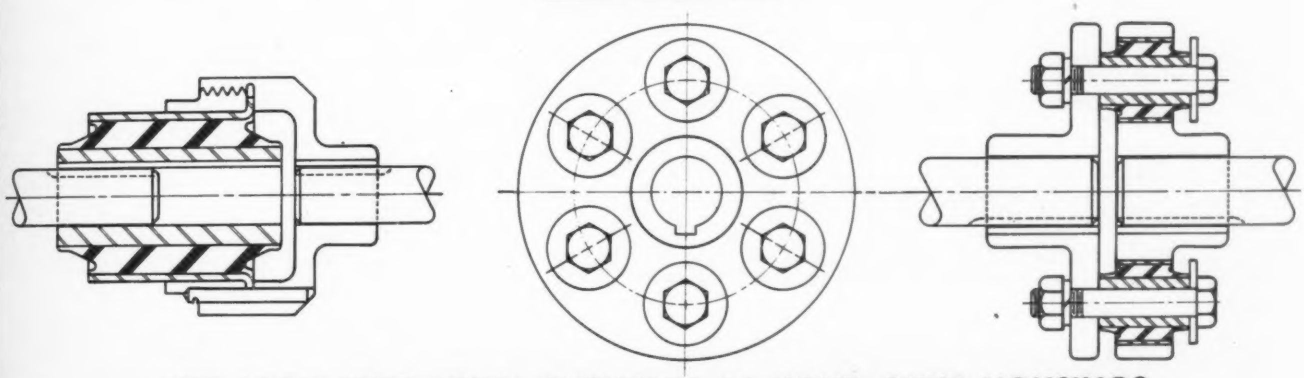
Eliminate { VIBRATION / NOISE / LUBRICATION }

MULTIPLE H. P.

The cross-section view below illustrates a single Lord Flanged Bushing used as a flexible coupling. Lord Bushings of various sizes can be used in this design to suit many multiple H.P. transmission requirements.



A typical compound application of Lord Bushings in a coupling is shown in the cross-section below. The number of bushings required depends on power to be transmitted and dimensional space limitations.



USE LORD TUBE FORM BUSHINGS IN YOUR OWN HOUSINGS

LORD Bonded Rubber Bushings can be utilized in many different designs to provide a simple and efficient method of flexibly connecting shafts, and particularly where parallel or angular misalignment exists.

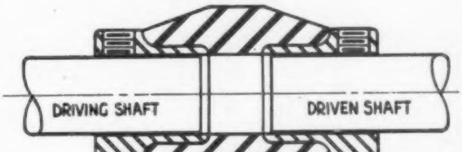
For constructing couplings to your own design, standard Lord Tube Form Bushings are recommended on multiple horsepower drives. Typical single and compound applications are illustrated above. Lord Bushings are made in a wide range of sizes in both flanged and straight tube styles to fit any dimensional condition or power requirement.

For fractional horsepower drives, Lord manufactures one-piece Bonded Rubber Couplings, shown at the right, in seven sizes with capacities from 1/16 H.P. to 1 H.P. at 1750 R.P.M.

Long operating life is built into Lord Bushings and Couplings through the use of special rubber flexing compounds and an exclusive method of bonding rubber to metal. The proper use of these Lord products results in minimum power loss, and prevents the transfer of vibration and noise along the shaft. No lubrication is required in this coupling design, as the metal parts are separated by and bonded to the rubber element.

Literature is available on both Lord Tube Form Bushings and Flexible Couplings. A letter outlining your design requirements will receive prompt attention.

FRACTIONAL H. P.



USE LORD COUPLINGS

	H.P. at 1750 R.P.M.						
	1/16	1/8	1/4	1/3	1/2	3/4	1
O.D.	13/16"	1"	1¼"	1⅜"	1⅝"	1 13/16"	2
Length	1⅜"	1¾"	2⅛"	2¼"	2½"	2 11/16"	2⅞
Std. Bore*	¼"	⅜"	½"	⅝"	¾"	¾"	¾

*Bores other than standard can be supplied.

LORD MANUFACTURING COMPANY . . . ERIE, PA.

245 E. OLIVE AVE., BURBANK, CAL. 280 MADISON AVE., NEW YORK 844 N. RUSH ST., CHICAGO

IT TAKES RUBBER IN SHEAR TO ABSORB VIBRATION

Crusher Poses Severe Drive Problems

(*Continued from Page* 116)

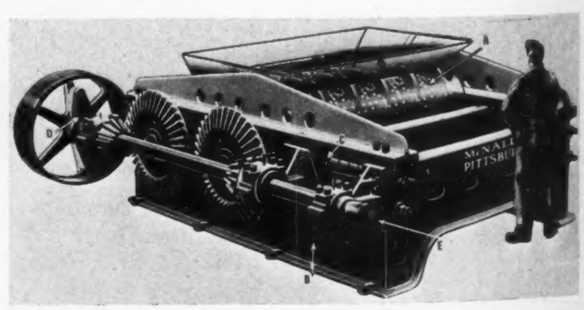
from any installation of such machines.

Double-roll breakers, such as shown in the illustration, rely on the impact of special teeth for the bulk of the reducing. Shown at *A*, these teeth, which may be cast iron or cast steel, depending on the hardness of the coal, are designed to crack down the large lumps of coal fed to the machine and to size them between 6 to 20-inch cube size with a full adjustment range within these figures. The design of the teeth and rolls is such that fines are not noticeably increased over the feed percentage. In fact, it has been found in operative experience that to screen out the small sizes before breaking adds approximately the same percentage of fines as where the entire feed passes through the breaker.

Protected by Electric Controls

One of the most important design features of the double-roll breaker is adjustment of the crushing rolls. This manual and reversible adjustment, *B*, is dependent on the rack and pinion mounted on each side of the one movable roll. This roll is on sliding bearings and while the bevel geared rollshaft moves, the bevel gear which drives the movable roll slides along the splined jackshaft in fixed relation with the bevel geared rollshaft. Adjustment may be made during operation to prevent the annoyance of draining coal out of the plant merely to set the breaker.

Of equal importance in the design is safety of operation. Raw coal may contain stone or other foreign material such as tramp iron. Along with these materials may come an unexpected feed surge of abnormal tonnage. The breaker must



clear itself of these hazards or motors may burn out and teeth break. For this reason all materials chosen for the crusher construction must be planned to give an extreme safety margin of stress allowance. However, there are safety devices which reduce the machine's dependency on sheer weight and rugged construction to prevent damage. First are the relief springs, *C*, which allow the movable roll to move back slightly should some abnormal condition threaten the rolls. Second, there is a

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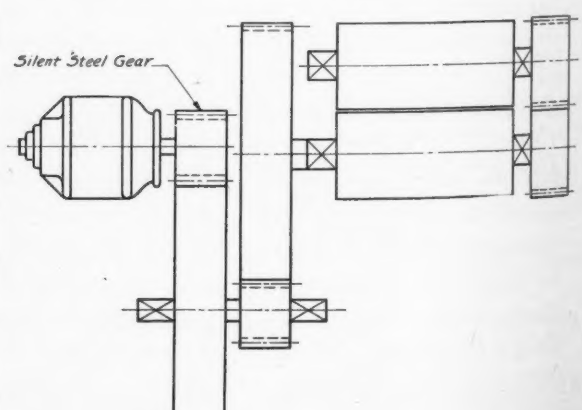
safety hub, *D*, which connects the main drive sheave to the driveshaft. This is secured to the sheave by hard maple dowels which will shear if an excessive strain is thrown on the crushing rolls. Third, and last, is an interceptor switch, *E*. This is the best insurance possible, as it protects not only the breaker but also the feeding and collecting apparatus. Installed on the main shaft, the switch is interlocked electrically with the feeder ahead of the breaker or other delivering machinery. Should the breaker choke or stall, the shaft revolutions per minute drop below the switch limit and the breaker motor cuts out. Simultaneously, interlock relays throw out the feeder motor, preventing a pile-up of raw coal in the breaker hopper. For further protection the interlock is so arranged that the feeder is stopped automatically when the conveyor receiving coal from the breaker is stopped.

Silent Steel Gear Extends Drive Life

By John A. Clark
John Waldron Corp.

COMPLETE reliability and long life are primary requisites of machine drives in the present all-out war effort. Especially is this true in gear drives where the time lost in replacement may be great. Effective insurance against such failure has been attained, in the design of a Stowe-Woodward hard rubber dust grinder, by the use of a special silent steel gear.

In this severe application a 75-horsepower motor, running at 860 revolutions per minute, is used as the prime mover. The silent gear (19 teeth, 2½-



inch diametral pitch and 8-inch face) is mounted on the motor shaft as indicated by the illustration. Previously, various types of pinions such as bronze and nonmetallics had been used with only in-

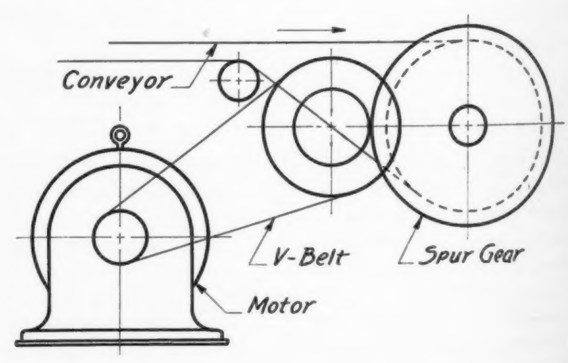
different success because of the high shock loads and the fact that the gear runs in oil.

The silent steel gear is constructed of steel laminations coated with minute particles of graphite. These laminations are compressed under hydraulic pressure and held securely by a circle of steel rivets. The lubricated laminated construction prevents noise vibrations from being transmitted from one lamination to the adjacent ones, resulting in mechanical flexibility and deadening of sound. Their greater strength makes possible the use of a smaller diameter or narrower face, resulting in a more compact drive. Not dependent upon deformation of the tooth profile for quiet operation, the pinion maintains true rolling engagement with its mating gear.

War Necessitates Use of Alternative Parts

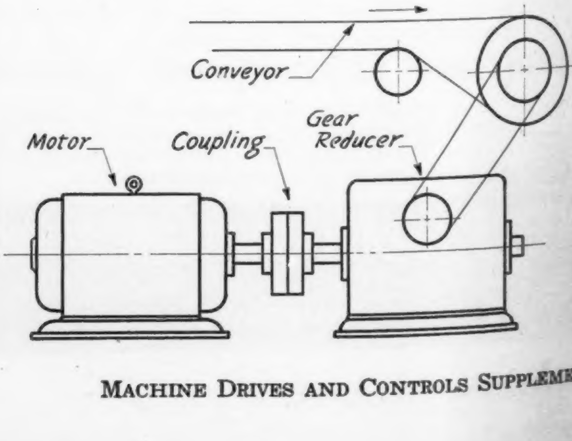
By H. C. Keller
Engineering Manager
Lamson Corp.

CONVERSION of old, and building of new plants for armament production have greatly stimulated the demand for mechanical conveyors. As designed and manufactured by the Lamson Corp., such conveyors have been customarily powered by gear-head motors. Coincident



with expanded volume, the ready availability of these prime movers has been reduced, with the result that alternative designs have had to be developed.

The problem of replacing the gear-head motor



without material sacrifice of economy and compactness and with a minimum increase in complication and number of parts is of first order magnitude. The most important of the developed alternatives are two in number.

Consisting of a motor and standard gear reducer, flexibly coupled, the first is comparable in efficiency but requires more space. The second comprises a completely designed drive unit, using spur gears and V-belts to obtain the necessary reduction and torque from the electric motor.

Urgency of the war effort more than justifies the minor sacrifice involved in hastening the day when all armament plants will be producing at full speed. With emphasis on mass production, conveyor systems play an important part in the success of this effort.

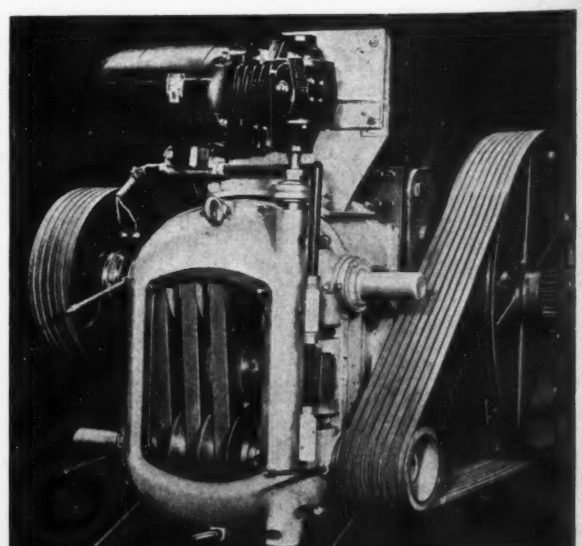
Pilot Control Regulates Strip Tension

By F. J. Donahue

Allis Chalmers Mfg. Co.

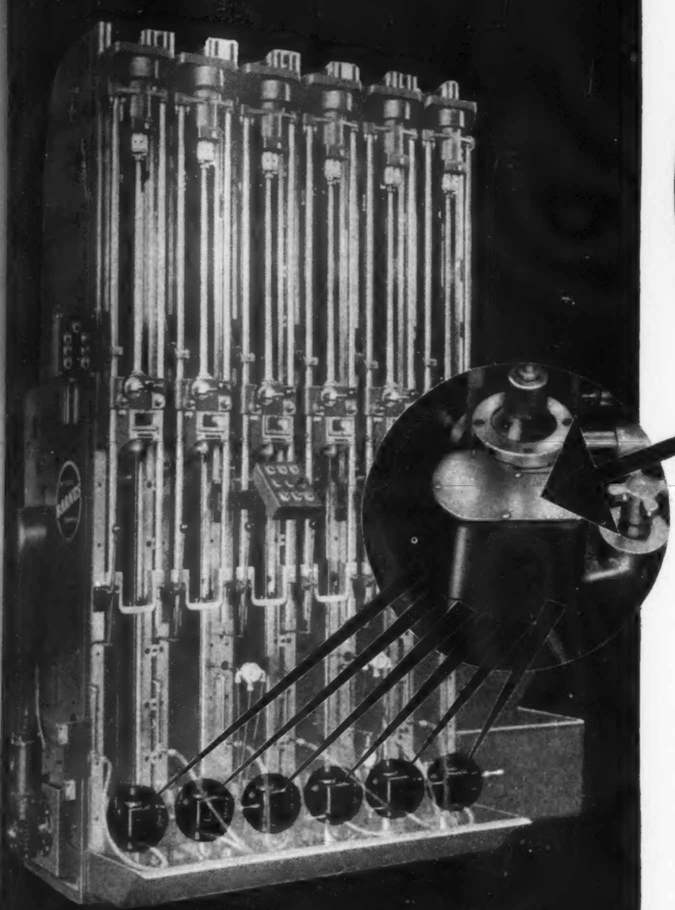
IN BRASS and steel mills, thin metal strip as it comes from the rolls is wound on a reel or blocker. To prevent stretching or tearing of the strip, substantially constant tension must be maintained between the reel and the rolls. Obviously, as more and more metal is wound onto the reel its diameter increases and in order to maintain constant strip speed and tension some form of speed control is necessary.

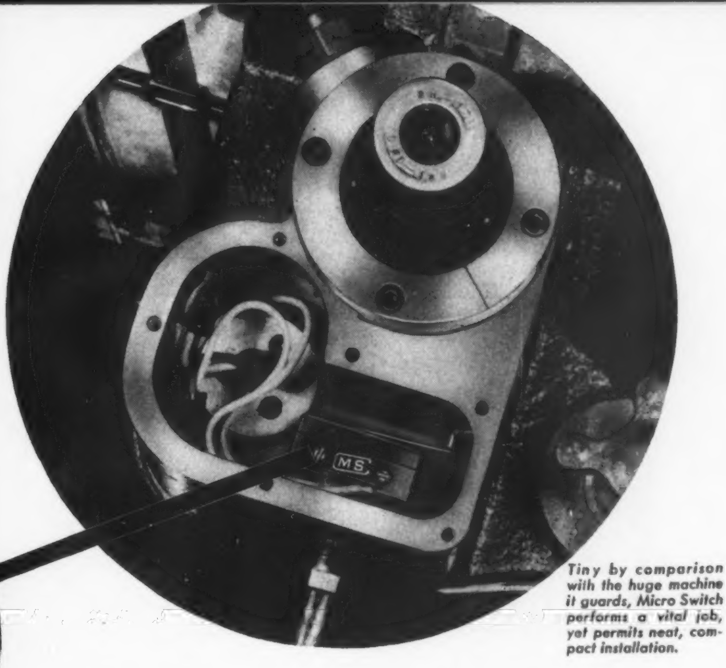
This problem is effectively solved by use of a torque sheave in conjunction with a variable pitch



speed changer. The torque sheave consists essentially of a free moving rim riding on a fixed hub. This hub is keyed to the input shaft of the speed changer.

Rim and hub of the torque sheave are linked





Tiny by comparison with the huge machine it guards, Micro Switch performs a vital job, yet permits neat, compact installation.

When rifle barrels are drilled . . .

MICRO SWITCH STANDS GUARD

At each Spindle of this W. F. and John Barnes 6-Spindle Vertical Rifle Barrel Drilling Machine, a Micro Switch Protects Expensive Drills from Damage . . . helps prevent costly delays

THIS towering machine conserves floor space and brings the automotive technique to the drilling of gun barrels. Each spindle operates independently, and each is guarded against torque overload caused by dull drills or clogging of chips by a tiny Micro Switch near its base. Here, as in hundreds of other applications, Micro Switch dependability helps prevent material damage and time loss . . . helps keep America's war production climbing.

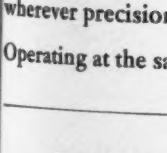
Almost countless are the applications of Micro Switch in industry. Small size, precise action, and absolute dependability have earned Micro Switch a unique kind of recognition wherever precision snap-action switches may be applied.

Operating at the same point through millions of operations . . . actuated by minute changes of movement or energy—Micro Switch is helping to solve hundreds of otherwise difficult design problems.

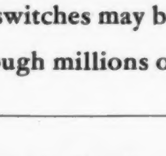
The basic Micro Switch is thumb-size—measures only 11/16″ x 27/32″ x 1 15/16″, weighs only an ounce, operates on pressures as low as 1¼ ounces and movements as low as .0002″. Listed by Underwriters' Laboratories with ratings of 1200 V.A. loads, from 125 to 600 Volts A.C.

Micro Switch can be supplied in a wide range of types and housings to meet a variety of operating conditions and requirements. The experience and skill of Micro Switch engineers is available to help you find the best possible answer to precision switching problems.

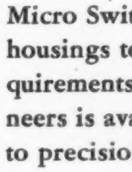
Other Typical Micro Switch Applications



LATHE CARRIAGE STOP
Lathe carriage actuates the Metal Clad Micro Switch at end of travel.



LEVER CONTROL
Deflection of lever by dog or cam operates roller arm of precision Limit Micro Switch.



Every engineer should have a copy of this informative Micro Switch catalog. The operating principles and applications of precision snap-action switches are pictured and described in detail. Ask for your free copy of Catalog No. 60.

READY SOON—Aircraft and similar applications of Micro Switches covered in new Catalog No. 70.

Micro Switch is a trade name indicating manufacture by Micro Switch Corporation.

MICRO MS SWITCH

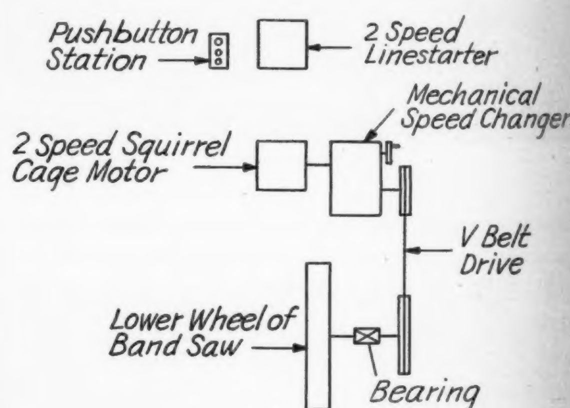
Manufactured in FREEPORT, Illinois, by Micro Switch Corporation. Sales Offices: New York, Chicago, Boston

through a compression spring, so that an increase of torque or pull of the belts on the grooved rim compresses the spring and produces relative movement between rim and hub. This movement actuates a sensitive switch which causes the pilot motor (electrically controlling the speed-changer) to slow down the blocker, thus maintaining a constant strip speed. The torque sheave can be adjusted while in motion to provide for various tensions.

A-C Drive Simplifies Speed Control

By L. W. Herchenroeder
Westinghouse Electric & Mfg. Co.

METAL cutting band saws require a wide speed range drive in order to use the saw blade at the most efficient speed for different kinds of material. This application is a "natural" for the alternating current adjustable speed drive because of its low cost, 10 to 1 speed range, simplicity and convenience. The drive consists of a unit frame motor-generator set, a two-



horsepower gearmotor, a magnetic linestarter with a two button pushbutton station, and a speed adjusting rheostat. No exciter is needed.

One of the features of this drive is the mounting of the lower wheel of the band saw directly on the shaft extension of the gearmotor, thus eliminating the extra bearings and coupling that otherwise would be required. The motor-generator set, being of the unit frame construction, has no bedplate and may be installed as easily as an ordinary motor at any convenient place, usually at the side of the band saw. The entire speed range is obtained by a single rheostat mounted on the frame of the band saw where it can be conveniently adjusted by the operator.

Overload protection for the drive is provided by the standard overload relays mounted in the linestarter. In addition a thermoguard is used on the direct-current driving motor. This protects the motor in case of overheating due to operation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The continuous wedging action of the De Laval worm gear transmits power without vibration, shock or noise. • Three or more teeth are always in contact, resulting in an even flow of torque and great strength. • Both motor and driven machine are saved the impact of gear or sprocket teeth. Frequently the product is improved or the machine can be speeded up because of the absolutely uniform motion. Operatives do more and better work when not annoyed by noise and vibration. Describe requirements and ask for Publication W-1129.
Double reduction worm gear with vertical output shaft.
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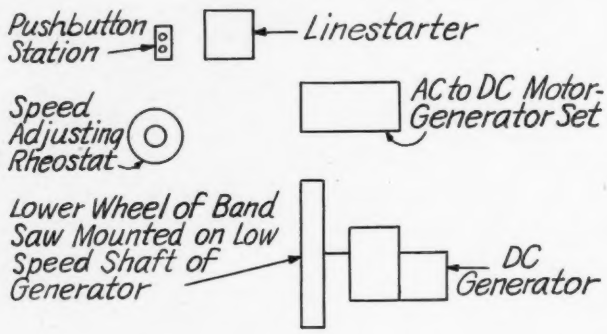
at excessively low speeds and continued overload.

This drive has several advantages over the combination of a two-speed squirrel cage motor and a mechanical speed changer. The mechanical design of the band saw is considerably simplified because no bearing, shaft or sheave is required for the lower wheel. The belt drive and mechanical



speed changer are eliminated which decreases maintenance, saves space, and improves the appearance of the machine. The operation of the machine is more convenient as the entire speed range may be obtained by adjusting one conveniently located rheostat instead of having several adjustments as is necessary with other types of drive. A comparison of the first and second illustration indicates the extent of the simplification achieved. Absence of any mechanical interconnection between units of the controller promotes flexibility in installation.

Streamlined for Protection

EVIDENCE that the airplane industry has plenty of company in the progressive development of military models is provided by the illustrated medium tank built by the American Locomotive Co. In contrast with the sharp corners and riveted upper and lower hull of the earlier model this new development is cast steel, characterized

by rounded contours capable of deflecting enemy shells. Increased speed of assembly and lessened machining requirements provide the combined additional advantages of faster production and the conservation of machine tool facilities for other much needed armament equipment.

MACHINE DRIVES AND CONTROLS SUPPLEMENT

APRIL 1942

CONTENTS

THE SUPREME TEST

MACHINE DESIGN *Editorial*

Stabilization—Not Stagnation!

AS KNOWN by every alert machinery manufacturer, design is never static. Dismal failure has attended the efforts of some builders in peacetime to freeze their designs for too long a period. Competition plays havoc with a producer who, feeling he has virtually a monopoly, clamps down on design changes and improvements because his meager knowledge of market conditions has blinded him to realities.

A happy medium in this connection is struck by the machinery builder who stabilizes his design just long enough to enable him to keep ahead of competition and at the same time put into production sufficiently large quantities to realize a profit. Few better examples of this action could be cited than the automobile industry, foreseeing market requirements yet stabilizing in most cases on a new model each year.

To meet the demands of the War Production Board for "production now", such stabilization has already taken place in the design of American armament equipment and the myriad types of machines for producing and supporting it. Rate of production will depend to a considerable extent on the job designers have done thus far in developing equipment that can be produced on a scale that not even labor stoppages, shortage of skilled manpower, restrictions on materials or other delaying factors can seriously affect.

It is that kind of production which will go a long way toward victory—only such design changes being made as will enable manufacture to go ahead even faster. Then will come the need for new "models". Reports of actual performance on the field of battle, supplemented by the embryo designs already on the board, should give every opportunity for design ingenuity to push American equipment way out in front. The present plans of the country's largest builders of armaments to place engineer-observers near the battle fronts is clearly a step in this direction.

Production now, with design development for even faster and better production later, is the order of the day. Initially, the responsibility for ultimate success rests with the designer. If the designs thus far produced could not be turned out fast enough, or if new developments did not far exceed those of the Axis in performance, the war might well be lost.

But those things are not happening, nor will they. The god of war depicted in the cartoon on the preceding page wants to know how good we really are in a war of machines. American engineers will show him!

L. E. Jermy

Designing Mountings for Rubber Under Shear

By Elmer Latshaw
J. G. Brill Co.

NEED for a simplified method of designing rubber springs has prompted the development of nine sets of formulas covering nine characteristic types of rubber mountings. These will be presented in data sheet form in two parts, beginning in this issue on Page 175. To facilitate design calculations the various formulas have been simplified to close approximations which, however, should easily satisfy the requirements of most practical engineering applications

ALL of the springs considered in the following involve rubber stressed in shear, this being the desirable mode of stressing to give large deflections. The rubber is bonded to steel plates or tubes which serve to mount the spring and deliver the load to the rubber area. Such a spring construction will guide the supported load in a frictionless manner.

Since the energy that can be stored in any spring varies as the square of the stress, it is desirable to work to the highest value which will give satisfactory life. Low stress gives low efficiency, making the unit heavy, large in size, and more expensive.

Parallel shear units like *Figs.* 1, 2, 3 and 4, carrying a load continuously, should be limited to 25 pounds per square inch based on the static weight supported. If the load varies, the limit should be 35 pounds per square inch based on the maximum static load.

For torsional units like *Figs.* 5, 6, 7, 8 and 9, subjected to a continuous torque, a good limit is 50 pounds per square inch. If the torque varies widely as in an automotive driveshaft, then the limit is 70 pounds per square inch based on the maximum torque that can be developed. Torsional springs can be stressed higher than the parallel shear type because the bond area under shear is continuous, unbroken and free of stress concentration.

Energy Varies Inversely as Modulus

The energy stored in a spring also varies as the reciprocal of the shear modulus. Approximate relations between rubber hardness and shear modulus are given in Table I.

This tabulation covers the practical range of rubber spring grades, 30 durometer being the softest commercial stock; the high modulus for 60 durometer establishes this as an upper limit. In the interest of high efficiency, the softer grades are preferable. When preparing a design for a new applica-

tion it is safer to make use of 40 to 45 durometer hardness; then, should the deflection be insufficient, a softer grade could be substituted and complete redesign of the spring or machine avoided.

A variation of plus or minus 5 per cent in the deflection or shear modulus is to be expected. The rubber manufacturer should be given time to experiment with compounding and cure techniques for new designs, otherwise greater variations may be expected on the first pieces made. Working drawings should specify the nominal durometer hardness, but the tolerance should be specified on the deflection only. Include on the drawings a complete description of the loading conditions; also give tolerances on all dimensions involved in mounting the spring on the machine.

TABLE I

Rubber Hardness vs. Shear Modulus

Hardness (Shore Durometer)	Shear Modulus (lbs. per sq. in.)
30	50
40	70
50	95
60	140

Thickness of the steel plates to which the rubber is bonded will be determined by end-load bearing area and the bearing value for mounting bolts. Plates must be heavy enough to avoid distortion or flexing due to transmitting the load to the rubber, and to counteract rapid deterioration by rust or other corrosive attack.

Thickness or least dimension of the rubber should not exceed 2 inches, otherwise difficulty may be experienced in accomplishing uniform and complete cure throughout the volume. If the required deflection calls for greater thickness than 2 inches, two or more units can be used in series.

Protect from High Temperatures

High temperature and oil have a deteriorating effect on rubber. Beyond 120 degrees Fahr. the weakening effect of temperature becomes noticeable and 150 degrees Fahr. must not be exceeded. Oil attacks rubber, making it soft, sticky and weak, so protection against oil is essential. A rubber spring unit should give from 5 to 10 years life with proper stress limits and protection from high temperature and oil.

Rubber is a good electrical insulator. When electrical machines are supported on rubber springs it may be necessary to provide flexible shunts for insuring personal safety or for completing the electrical ground circuit of the machine. This should always be investigated to obviate trouble due to electric currents following unintended paths and damaging bearings or other parts.

A similar set of design equations is given for each type of spring and numbered alike for reference. Equation 1 serves to define

Nomenclature

A = Rubber bond area, square inches
D = Deflection, inches
E = $PD = 2\times$ work stored, inch pounds
e = Base of natural logarithms
G = Shear modulus of rubber, pounds per square inch.
H = Height, inches
H' = Height, inches
h = Height (variable) inches
k = Radius factor, pure number
m = Function of k, pure number
n = Function of k, pure number
P = Load, pounds
q = Shear stress, pounds per square inch
r = Radius, inches.
R = Radius of torque, inches
t = Rubber thickness, inches
θ = Torsional deflection, degrees
V = Rubber volume, cubic inches
x = Variable radius, inches

the shape of the rubber, this being important in certain cases. Other equations are listed in the order usually required when preparing a design. Equation 10 is included to give an idea of relative efficiency for each type. The energy stored is dependent on the volume of rubber, stress squared and shear modulus; also a term m/n the value of which depends on how uniformly the volume of rubber is stressed. If the m/n term is not present in Equation 10 the stress distribution is uniform, or m/n is equal to unity.

Design Data for Rubber Mountings

Part I

(2) $A=\dfrac{P}{q}$

(4) $t=\dfrac{DG}{q}$

(5) $q=\dfrac{DG}{t}$

(6) $D=\dfrac{Pt}{AG}$

(7) $P=Aq$

(8) $A=$ Depends on geometric shape in width and height

(9) $V=At$

(10) $E=\dfrac{q^2V}{G}$

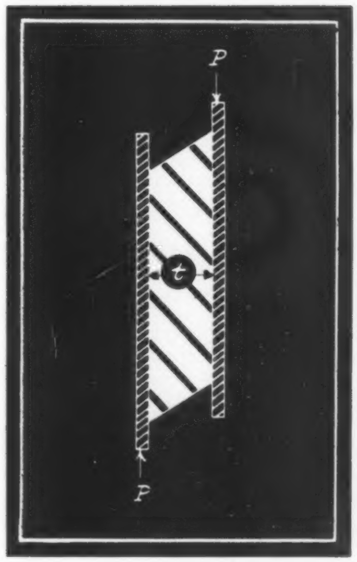


Fig. 1—Representing the general case consisting of a flat slab of rubber bonded to parallel steel plates, the area can be of any suitable shape such as a square, rectangle or circle

(1) $xh=rH$

(2) $rH=\dfrac{P}{2\pi q}$

(3) $m=\dfrac{DG}{rq}$

(4) $t=\dfrac{DG}{q}$

(5) $q=\dfrac{DG}{t}$

(6) $D=\dfrac{Pt}{AG}$

(7) $P=Aq$

(8) $A=2\pi rH$

(9) $V=At$

(10) $E=\dfrac{q^2V}{G}$

(11) $m=(k-1)$

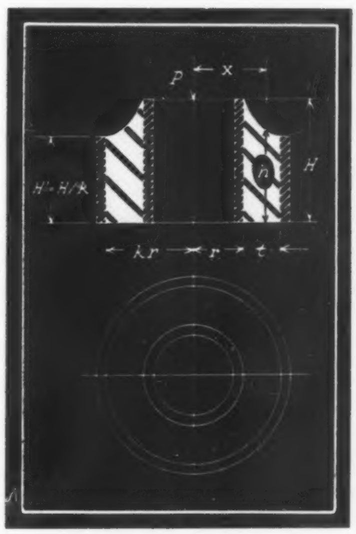


Fig. 2—Height of the rubber is proportioned to give uniform area in shear at any radii, hence the unit stress is uniform throughout the rubber and at the bond surface on both metal tubes

(1) $H'k = H$

(2) $rH = \dfrac{P}{2\pi q}$

(3) $m = \dfrac{DG}{rq}$

(4) $t = (k-1)r$

(5) $q = \dfrac{DG}{rm}$

(6) $D = \dfrac{Prm}{AG}$

(7) $P = Aq$

(8) $A = 2\pi rH$

(9) $V = Arn$

(10) $E = \dfrac{q^2V}{G} \; \dfrac{m}{n}$

(11) $m = \dfrac{2k \log_e k}{k+1}$

(12) $n = \dfrac{(k^3+3k^2-3k-1)}{6k}$

k-l	m	n	m/n
.0	.000	.000	1.00
.2	.199	.201	.99
.4	.392	.408	.96
.6	.578	.623	.93
.8	.756	.847	.89
1.0	.924	1.083	.85
1.2	1.083	1.331	.82
1.4	1.235	1.591	.78
1.6	1.381	1.863	.74
1.8	1.518	2.147	.71
2.0	1.649	2.444	.68

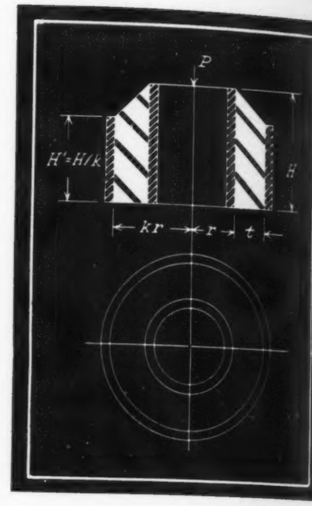


Fig. 3—Bond areas on both metal tubes are equal. The edge of the rubber follows a straight line and the excess of rubber gives lower shear stress in the rubber at intermediate radii

(1) $H' = H$

(2) $rH = \dfrac{P}{2\pi q}$

(3) $m = \dfrac{DG}{rq}$

(4) $t = (k-1)r$

(5) $q = \dfrac{DG}{rm}$

(6) $D = \dfrac{Prm}{AG}$

(7) $P = Aq$

(8) $A = 2\pi rH$

(9) $V = Arn$

(10) $E = \dfrac{q^2V}{G} \; \dfrac{m}{n}$

(11) $m = \log_e k$

(12) $n = \dfrac{k^2-1}{2}$

k-l	m	n	m/n
.0	.000	.00	1.00
.2	.182	.22	.83
.4	.336	.48	.70
.6	.470	.78	.60
.8	.588	1.12	.52
1.0	.693	1.50	.46
1.2	.788	1.92	.41
1.4	.875	2.38	.37
1.6	.956	2.88	.33
1.8	1.030	3.42	.30
2.0	1.099	4.00	.27

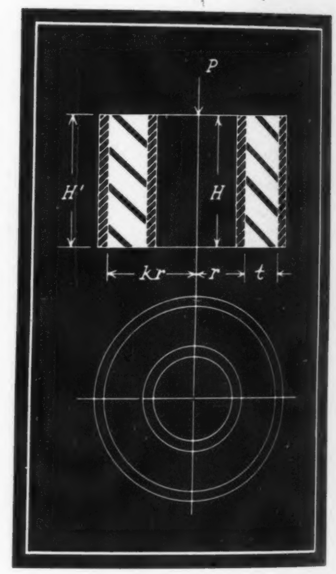


Fig. 4—Since the unit is of uniform height, the bond area on the inside tube is the smaller. Stress is greatest at the inner tube and diminishes toward the outer tube giving rather low efficiency

(1) $x^2h = r^2H$

(2) $r^2H = \dfrac{PR}{2\pi q}$

(3) $m = \dfrac{DG}{Rq}$

(4) $t = (k-1)r$

(5) $q = \dfrac{DG}{Rm}$

(6) $D = \dfrac{PR^2m}{ArG}$

(7) $P = \dfrac{Arq}{R}$

(8) $A = 2\pi rH$

(9) $V = Arm$

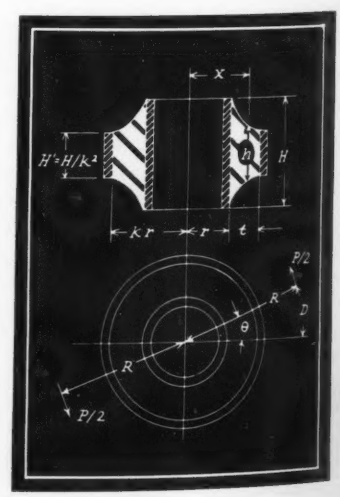
(10) $E = \dfrac{q^2V}{G}$

(11) $m = \log_e k$

(13) $\Theta = \dfrac{180\,D}{\pi R}$

k-l	m
.0	.000
.2	.182
.4	.336
.6	.470
.8	.588
1.0	.693
1.2	.788
1.4	.875
1.6	.956
1.8	1.030
2.0	1.099

Fig. 5—Height of the rubber is proportioned to give uniform shear stress at any radii in the rubber and at the bond areas on both tubes



To the MANUFACTURER with VISION for TOMORROW

Technical information regarding all types of SLEEVE BEARINGS such as Alloys, Lubrication, Design.

Research to determine the CORRECT answer to unusual bearing applications.

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War has interrupted normal business but it hasn't stopped thinking . . . and planning for tomorrow. Leading manufacturers in every line of products are now seeking new ways to improve . . . to cut costs . . . to increase performance. All this is possible with the *right* type of bearings.

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JOHNSON BRONZE COMPANY

Sleeve BEARING HEADQUARTERS

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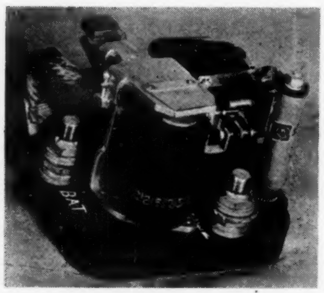
New PARTS AND MATERIALS

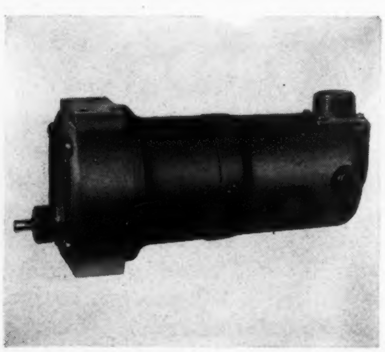
Aircraft Reverse Current Cutout

WEIGHING less than two pounds and performing the same function as an automobile cut out, a new aircraft reverse current cutout has been announced by Westinghouse Electric & Mfg. Co., East Pittsburgh, Pa. It permits the flow of generator current to the storage batteries when the generator voltage is great enough to charge the batteries, but disconnects the two units when the voltage drops below the charging point. By thus breaking the circuit, the cutout avoids wasteful reverse current which would flow out of the battery and diminish the small store of power. Added advantages are greater accuracy, more current capacity and special construction to withstand shocks and vibration caused by the powerful main engines.



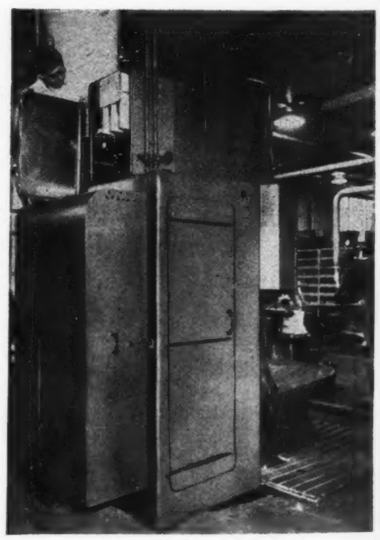
Aircraft Motor Announced

WITH a horsepower rating of 1/16 at 1500 revolutions per minute, the E2Y1PC aircraft motor of Dumore Co., Fourteenth and Racine streets, Racine, Wis., incorporates a magnetic clutch and brake in its design. Clutch disengages armature from gears at the instant the circuit is opened and the brake brings the output shaft to a stop in a fraction of a second. This motor is available in twelve modifications. It can be supplied with the basic motor only; basic motor with clutch; basic motor with single or double gear reductions; and basic motor with single or double gear reductions and magnetic clutch. Overall length of the motor is 8 inches; weight, 4 pounds. Other available ratings are ⅛ horsepower at 1500 revolutions per minute, 1/10 at 200 revolutions per minute, and ⅛ horsepower at 200 revolutions per minute. Motors are now used for applications such as cowl and wing flap controls, oil cooler controls, and aerial camera controls.



Air Circuit Breakers

IN ACCORDANCE with the present trend of incorporating, as integral parts of resistance welding machines, devices for protecting, controlling and timing, General Electric Co., Schenectady, N. Y., has brought out its new line of air circuit breakers. Those offered for this purpose are designated as Types AE-1AY1 and AE-1BY1. They are metal-enclosed, built to fit behind the steel panels of the welder, and can be removed easily for inspection by unscrewing two screws in the breaker case. Power is entirely disconnected from breaker when removed. Breakers have a wide range of calibration, offering choice of trip settings which provide full protection and allow maximum output of tube. The unit can be arranged to open the cir-



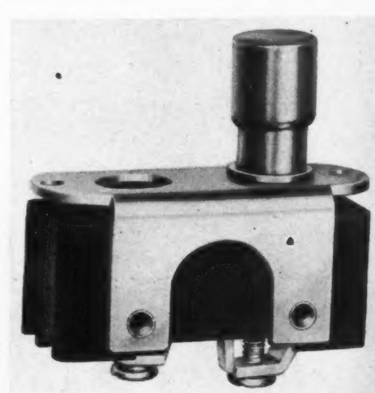
cuit in case of undesirable conditions such as "low-voltage" or no "water flow," and also to close the circuit remotely by means of a pushbutton.

Custom Made Lock Washers

OVER 100 custom-made lock or thrust washers are being produced by Dayton Rogers Mfg. Co., 2830 South Thirteenth street, Minneapolis, under a special die stamping process, at a comparative cost of about five or six hand-made pieces. This process assures absolute duplication throughout the entire lot. The washers are available in sizes of ½ to 12 inches in diameter of any sheet alloy that will lend itself to conventional die stamping of sheet metal parts. Washers are stamped on special compound piercing and blanking die assuring duplication of all blanks and pierced holes and slots.

New Switch for Aircraft

DESIGNED to meet the requirements of U. S. Air Corps specification 94-32249, the Mu-Switch Corp., Canton, Mass., announced recently its new switch, with wide contact spacing mounted in a cadmium-plated steel bracket to which is attached an over-travel plunger mechanism having a pretravel of 1/16-inch minimum and an over-travel of ⅛-inch, plus or minus 1/32-inch. One pound minimum, and eight pounds maximum pres-



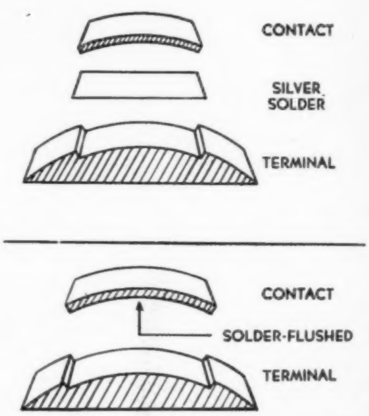
sure is required to operate the switch over the entire operating range. Electrical rating of switch is 25 amperes at 28 volts direct current, 17 amperes at 125 volts alternating current, and 8.5 amperes at 250 volts alternating current, noninductive loads. Wiring connections can be supplied normally open,

ZINC IN WAR
16 FUZE PLUGS EVERY 12 SECONDS!
DIE CASTING 6
METAL SPRAYING
GALVANIZING
PHARMACEUTICALS
NICKEL SILVER
HULL PLATES 1
RUBBER 2
PAINT 3
BRASS 4
CERAMICS 5
Any fabrication process which is capable of turning out 16 semi-finished metal parts every 12 seconds is geared to war production. That is why the die casting industry is now taking on the production of direct and indirect implements of war. That is why zinc alloy must be available in sufficient quantities to enable the die casting industry to make an all-out war effort.
Zinc alloy is the most widely used metal in die casting because, with the selection of the most appropriate alloy, it enables the production of castings having four outstanding characteristics: Accuracy—close tolerances can be achieved and maintained throughout the useful life of a casting; Complexity—intricate shapes are obtained to incorporate many features in a single part, thereby eliminating much machining and assembling; Strength—zinc alloy die castings have higher shock resistance than most other cast materials; Economy—all of the foregoing factors, plus long die life and low metal cost, add up to impressive production economies.
The die casting industry is doing a war production job (as typified by the shell fuze plugs shown in the background) and the zinc is there as needed. But this represents one more reason why it will be difficult for civilian users to obtain all of the zinc they would like to use.
THE NEW JERSEY ZINC COMPANY
MANUFACTURERS OF THE FAMOUS HORSE HEAD ZINC PRODUCTS
New Jersey Zinc

normally closed, and single-pole, double throw. Special mounting lugs are provided on bottom of switch to permit use of ring type wire connectors. Characteristics and dimensions of the over-travel plunger mechanism can be changed to adapt this new switch assembly to practically any type of application. This construction is superior inasmuch as over-plunger mechanism is mounted on the steel bracket instead of directly on the bakelite switch as before.

Contacts Coated with Silver Solder

BY THE use of solder-flushed contacts supplied by Gibson Electric Co., 8355 Frankstown avenue, Pittsburgh, considerable saving in production time when making electrical contact assemblies is possible. An operator in some cases can double the number of contact assemblies brazed in a given



time by using solder-flushed contacts. The contacts, coated on one side with silver solder, are applied immediately to the terminal and brazed in an electric welder, thus eliminating the extra operation of placing a separate piece of solder between the contact and the terminal. A uniform distribution of solder in the joint between the contact and the terminal is assured. Use of the solder-flushed contacts permits brazed joints to be used in contact assemblies. The solder-flushed electrical contacts can be supplied in practically all grades of the company's powdered metal contacts, and in silver and silver alloy contacts.

Direct-Connected Rotary Pumps

PLACED in production recently at the Blackmer Pump Co., Grand Rapids, Mich., is a new complete line of direct-connected rotary pumps known as the "D-C" line. These pumps are furnished in standard capacities of 5, 10 and 20 gallons per minute, and for pressures up to 100 pounds per square inch. At an operating speed of 1800 revolutions per minute the pumps deliver rated gallonage. To obtain lower capacity, motor speeds of 1200 or 900 revolutions per minute may be substituted. Pump is mounted on bedplate with flexible coupling

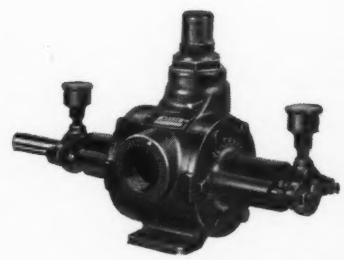
To aid users of Nickel alloys, thirty service centers are maintained in industrial areas. From these strategically located key points, our field representatives are on call to advise American industry about the selection, fabrication and uses of ferrous and non-ferrous materials. Assistance is also given on problems arising from the temporary lack of Nickel.

Through the years, research, field studies and user experience have all contributed to a fund of practical, time-proved information. Many of these data have been compiled in convenient printed form, useful both to experienced men handling new materials or performing unfamiliar operations . . . and to the many new employees.

Now . . . when minutes and materials are so vital . . . make full use of this metal-working experience. Send for a check list of helpful printed pieces on the selection, treatment, fabrication and use of Nickel alloys, or send your specific questions to:

THE INTERNATIONAL NICKEL COMPANY, INC. 67 WALL STREET NEW YORK, N. Y.

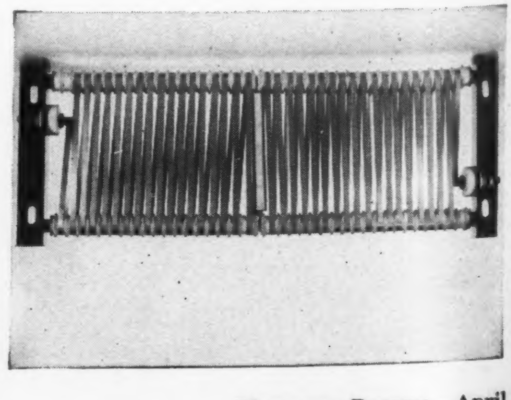
for direct connection to an electric motor, steam turbine or gasoline engine. Available also are pumps for belt drive complete with tight and loose pulleys and belt shifter. These pumps are furnished only for use as original equipment on various types of machines. Compact size of pump requires very small mounting space. Two large bearings, one on either side of rotor, are both designed for substantial overload and maximum shaft rigidity. Packing is ¼-inch square braided asbestos. Another feature of the bearing and gland construction is drain passage whereby vacuum is maintained on



the stuffing box, thus reducing leakage through gland. Pumps are self-priming and positive displacement, and are suitable for suction lifts of 27 feet, provided liquid does not vaporize under vacuum or pressure. They are not recommended for handling liquids having temperatures in excess of 250 degrees Fahr., but at lower temperatures will handle practically any clean liquid. Models DC-10 and DC-20 pumps can be furnished with built-in relief valves which are of ample size to by-pass entire capacities at low pressure settings, even when handling viscous liquids. DC-5 type pump can be supplied with a separate relief valve.

Air Heater Introduced

DESIGNED for temperatures of from 750 to 1150 degrees Fahr., Westinghouse Electric & Mfg. Co., East Pittsburgh, has brought out its new air heater for heating ovens or furnaces used in annealing of aluminum, glass and other materials. The unit has a heating element consisting of a one-piece nickel-chrome ribbon, spirally wound around heavy porcelain insulators. Rating is 5 kilowatts for operation on 220 volt circuits. Complete heater



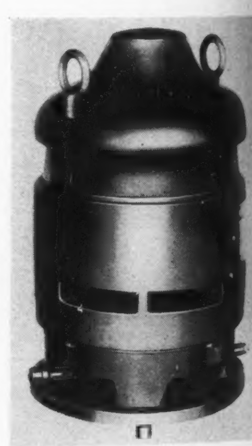
is 2½ inches thick and requires space of 12 x 33 inches for mounting.

Tubing for Electrical Insulation

DESIGNED for use as electrical insulation in the electronic, instruments, aircraft, electrical appliances and power industries, the new fibronized Koroseal tubing, developed by Irvington Varnish & Insulator Co., Irvington, N. J., has inside and outside smoothness, exceptional elasticity and close manufacturing tolerance. The tubing is resistant to acids, alkalis, solvents and heat, is fireproof, and possesses an insulation resistance of infinity after 16 hours at 90 per cent R.H. and 105 degrees Fahr. As a result of the A.S.T.M. Test of being subjected to 225 degrees Fahr. for approximately 1000 hours, it was found to have retained its flexibility. Tensile strength of the tubing is 2845 pounds per square inch, dry dielectric strength (.022-inch wall thickness) of 1050 VPM; wet dielectric strength (.022-inch wall thickness) of 817 VPM after 24 hours immersion. The tubing is available in all A.S.T.M. sizes and in a variety of colors as well as transparent.

Adds to Line of Motors

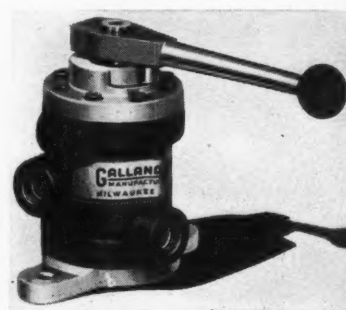
ADDED to the line of Tri-clad motors, made by General Electric Co., Schenectady, N. Y., are three new motors—a vertical general-purpose polyphase motor, a vertical shielded polyphase motor (1 to 20 horsepower), and a vertical shielded single-phase motor (1 to 5 horsepower). The general purpose motor with its compact, cast-iron frame and cover, is designed to give additional protection to electrical parts, without adding materially to overall height. Accurate alignment of motor to driven unit is assured by machined rabbets on the stator frame and base. Rotating weight and thrust are transmitted to base through lower bearing fitted to a one-piece base. Bearings are of deeply-grooved type, encased in machined cast-iron housings. Openings are shielded to bar entrance of chips and other objects. Grease fittings allow rapid lubrication without disassembly of motor. The polyphase shielded type motors are available in either solid-shaft or hollow-shaft construction, while the single-phase shielded type motors are available only in solid-shaft construction. Both of these are for normal-thrust or high-thrust applications. Air is discharged downward at low velocity, permit-



ting little or no dust disturbance around motor. These motors are suitable for outdoor operation except where extremes of moisture, dust, or other harmful agents make the selection of fan-cooled motors more economical. All frame sizes have common mounting dimensions, making possible interchangeability of many horsepower and speed combinations, including single-phase ratings. Motors can be rotated to any one of four base positions without modification of supporting structure.

Balanced Hydraulic Valve

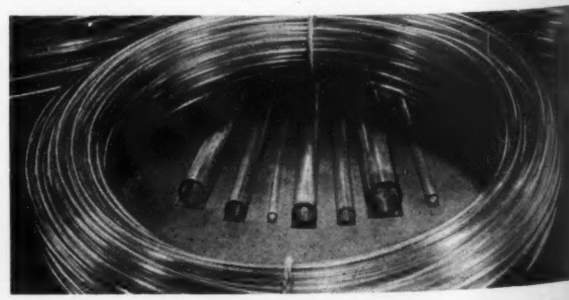
PROVIDING precision control of hydraulic pressures over 500 pounds per square inch, the new valve manufactured by Galland-Henning Mfg. Co., Milwaukee, overcomes obstacles such as strenuous and tiring valve-lever manipulation and tendency toward pressure-locking. Because the hydraulic



pressure inside of this new type valve is always balanced, it never becomes "pressure-locked." The lever can be operated easily from either the "on" or "neutral" positions, and as a result greatly steps up productivity of high-pressure hydraulic machinery and eliminates undue fatigue on the operator's part. There is only one moving part in the valve, and this fact together with the simplified design, makes it practically wearproof. The new valve is built in four sizes to fit most common sizes of hydraulic pressure lines, ⅜, ½, ¾ and 1-inch.

Extruded Seamless Tubing

TRANSPARENT seamless tubing is being extruded in continuous lengths by Extruded Plastics Inc., Norwalk, Conn., from a cellulose acetate butyrate formula of Tenite, produced by Tennessee



PRACTICAL DATA FOR PRESENT PROBLEMS

Here are two books designed to help users of Molybdenum steels and irons to conserve all alloying elements, and possibly steel and iron, by getting the most in the way of strength, toughness and wear resistance with the lowest alloy content.

"MOLYBDENUM IN STEEL" covers the fundamental metallurgy of Molybdenum steels. Heat treatment – physical properties – applications – of a number of these steels are treated at length.

"MOLYBDENUM IN CAST IRON" covers the effect of Molybdenum in gray iron, giving suggested analyses for practical applications and detailed discussion of high strength (60,000 p.s.i. and up) irons.

Both books will gladly be sent free on request.

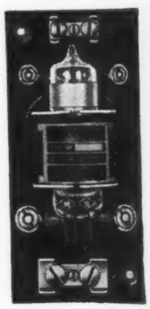
MOLY

Climax Molybdenum Company
500 Fifth Avenue · New York City

Eastman Corp., Kingsport, Tenn. This new plastic tubing is practically unbreakable and may be readily bent, formed or curved to fit any condition. Weld marks and joints are eliminated in the fabrication of tubing. Ends may be adapted easily to standard flared fittings with similar tools to those used for copper tubing. Large diameter tubing, with wall thickness of .0625-inch, can be threaded with standard thread-cutting tools. The tubing is available in sizes up to ½-inch in diameter, with wall thickness of .035-inch, in long length coils as shown in the accompanying illustration. Tubing over ½-inch in diameter is supplied in 12-foot lengths, while 1-inch diameter tubing is expected to be available for delivery shortly.

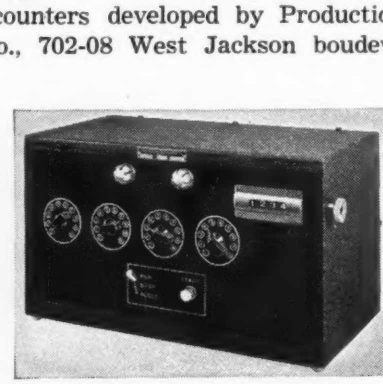
Silent Mercury Plunger Relay

COMPLETELY redesigned, the new mercury plunger relay has been improved by H-B Electric Co., 2530 North Broad street, Philadelphia. As a result of the improved design, the metal cap and support frame that held the mercury tube are eliminated, releasing strategic metals for the war. The mercury tube is made secure by a single metal band giving twice the stability of the previous design. Area for terminal fastening and connecting has been tripled, eliminating the danger of tube breakage and simplifying installation. The unit is available for either alternating or direct current and is rated at 30 amperes, 110 volts at 1 horsepower. According to the company, this improved silent mercury plunger relay has been tested to 10 million operations without failure. The relay has only one moving part.



Electric Predetermined Counter

EFFICIENT count controlling can be obtained through the electrically-operated predetermined counters developed by Production Instrument Co., 702-08 West Jackson boudevard, Chicago. Any number from 1 to 9999 can be quickly set up by simply turning knob pointers to proper digits. When the predetermined number is reached, a control and signal circuit is closed (or opened). This



circuit may be used to sound an alarm or operate a relay to perform any desired function. At the predetermined number, counting ceases until a reset lever is depressed, making the instrument ready for another count cycle. For use in counting objects delivered from a machine, the electrically-actuated counter may be located near to, or at a distance from, the things being counted. The counters are actuated by any switch, relay or photoelectric unit with a closed period of .035-second or more and an open period of .040-second or more.

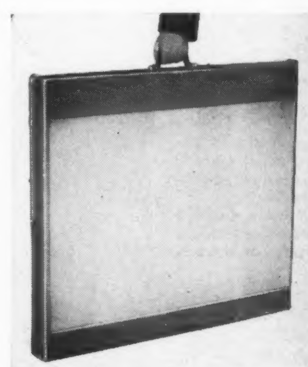
Low Visibility Paint

POSSESSING exceptional heat deflecting qualities, a new type of low visibility paint is being offered by The Arco Co., 7301 Bessemer, Cleveland, for use in protective concealment of vital defense equipment. The paint is being offered in green, tan, black and four intermediate shades which, when properly selected, will meet the requirements of good camouflage in any sort of terrain. All seven shades are hard to see from the air and throw off most of the heat of the sun's rays.

Engineering Dept. Equipment

Portable Tracing Board

AS a compact, space-saving tracing unit, Hamilton Mfg. Co., Two Rivers, Wis., has brought out its new portable tracing board No. 188 which can be used in any drafting room, on any drafting table. No extra space is needed for the portable tracing unit, and the draftsman can have several boards in use at the same time. Daylight fluorescent lamps furnish even, glareless light on the 24 x 36 inches plate glass tracing surface. Adjustable stop rods hold board in place when used on an inclined surface. Black, split-leather handle is bolted to top edge of board. Overall dimensions are 36 x 31 x 3⅜ inches. Board is made for 110 volt alternating current only.



Continuous Copying Machine

FOR making contact prints on subdued-light type photographic materials, the Paragon Revolute Corp., 77 South avenue, Rochester, N. Y., has designed a new continuous type copier known as Model 8F. It will make reproductions rapidly from any

type of original up to 44 inches in width by any length. Reproduced tracings may be made from blueprints, black and white or colored originals. Capable of copying originals which are either on transparent or opaque materials, printed on either one or two sides, this machine fills today's demand for a continuous type copier to make reproduced tracings and other copies. For instance, where head engineering offices make duplicate tracings for their branches and where subcontractors receiving one complete set of blueprints make many others the machine finds wide application. The principle is that of revolving contact, preventing slippage and blurred prints. An 8-inch diameter Pyrex glass cylinder is provided and original and sensitized material revolves with this cylinder. Three fluorescent lamps, two white and one gold, are located on the inside of this cylinder, the white being used for making reproductions from black

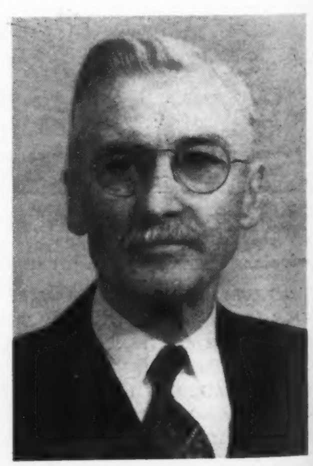
and white originals and the gold for reproductions from blueprints. Transparencies printed on one side are reproduced by transmitting light through them to sensitized material. Opaque originals, printed on two sides, are copied by reflecting or "bouncing" light back to the sensitized material. No special dark room, focusing or lenses are necessary.

Pencil with Plastic Ferrule

PLASTIC ferrules on lead pencils manufactured by Reliance Pencil Corp., Mt. Vernon, N. Y., have been substituted for brass ferrules on its "Quality" group, while such materials as paper-fiber, steel and wood are receiving consideration for the inexpensive line of pencils. The smooth appearance of the plastic-tip pencils is an improvement on the old style ridged, brass ferrule—proving the adage "Substitution offers the opportunity to improve." The material used is a cellulose-acetate formula, supplied by various plastic material producers.

MEN *Of Machines*

RECENTLY appointed head of the engineering department at Caterpillar Tractor Co., George E. Burks has had considerable experience in the design of heavy machinery. A native of Philipsburg, Mont., he obtained his engineering education at the Universities of Montana and California. His first design experience was his connection with Schmeiser Mfg. Co., manufacturers of land levelers and other tractor-drawn equipment. Mr. Burks joined the engineering staff of Western Harvester Co., then a subsidiary of Caterpillar, in 1928 and a year later was transferred to the parent company's engineering offices at San Leandro, Calif. In 1933 he was advanced from chief draftsman to supervision of the experimental and research engineering at the plant. Five years later he was moved to Peoria as assistant chief engineer in charge of engines.

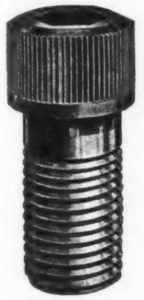


UPON recommendation of the Committee on Science and the Arts, of the Franklin Institute, Walter Larkin will receive a certificate of merit "in consideration of his admirable machine designing involving the ingenious application of known mechanical movements to the invention of circular knitting machines of special types." Senior inventor of Fidelity

Take Your Choice

There IS an UNBRAKO Socket Screw for Every Purpose . . .

Reg. U. S. Pat. Off.



U. S. & Foreign Pat's Pending
Fig. 1434
Knurled "Unbrako" Socket Head Cap Screw

Fig. 1663
Square Head Set Screw with Tool Post Point

Pat'd and Pat's Pending
Fig. 1646
"Unbrako" Self-Locking Square Head Set Screw

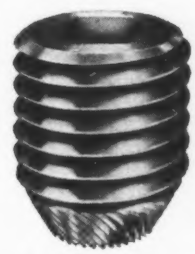


Fig. 1645—"Unbrako" Self-Locking Hollow Set Screw with Knurled Point

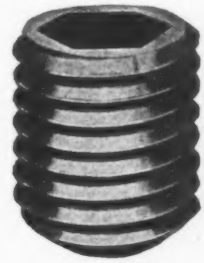


Fig. 232
"Unbrako" Hollow Set Screw

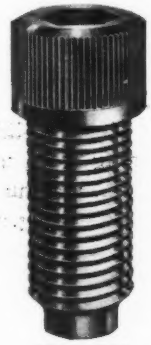


Fig. 1662
Knurled "Unbrako" Socket Head Cap Screw with Tool Post Point

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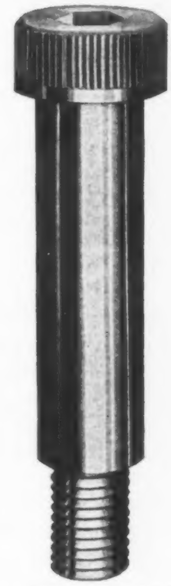

Pat. Applied for
Fig. 711—"Unbrako" Socket Head Stripper Bolt

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SPECIAL FILLISTER and other **HOLLOW HEAD SCREWS**

Fig. 1021-4

Fig. 1021-2

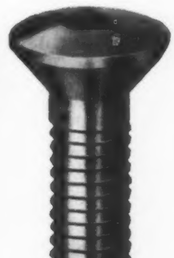


Fig. 1026-8

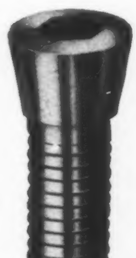


Fig. 1026-7

Fig. 1664

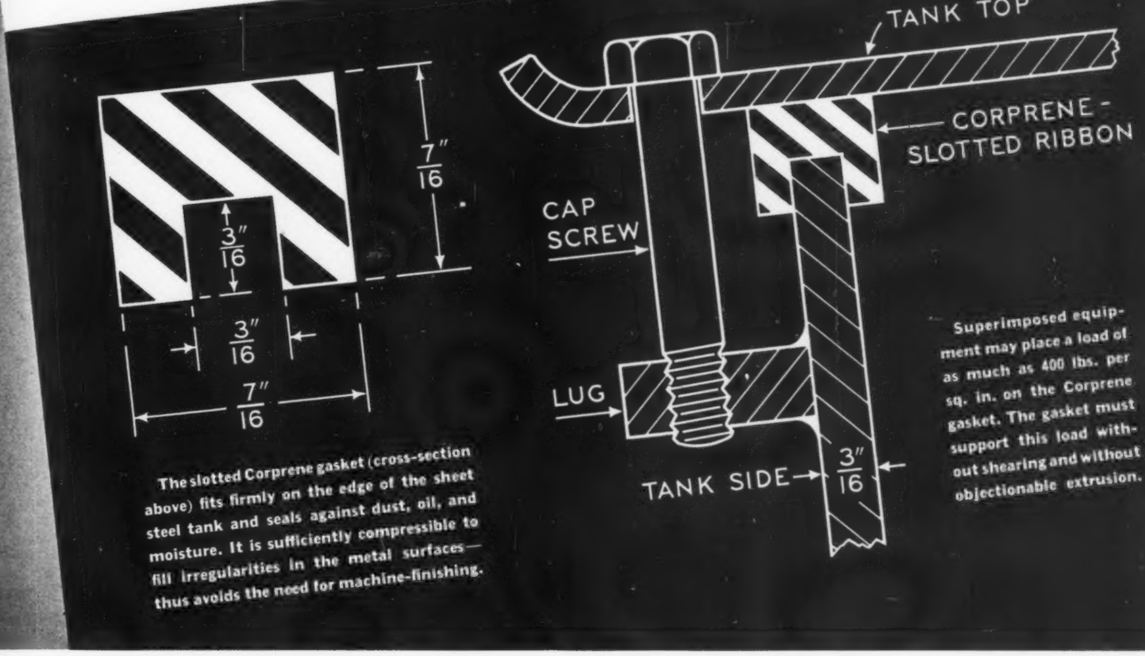
Machine Co., Philadelphia, he is well known to the textile industry and more specifically to that of the seamless knitting field. Spending his entire lifetime in this field, he has a total of twenty-seven inventions all of which have been assigned to the company. Mr. Larkin completed his education at Westtown school, Chester County, Pa., in 1896, and began learning the trade of pattern making. After completing his apprenticeship he became a draftsman and obtained a varied experience resulting in a broad mechanical and designing training among a number of old, well-established machinery building companies in Philadelphia. His first contact with knitting machine design was in 1905 when he joined Scott & Williams Co. Since that time to 1920 he was responsible for development of a great many improvements in circular knitting machines. As a designer he served with three of the five largest and most aggressive circular machinery builders. Mr. Larkin followed the footsteps of his father who was also an inventor.

ONE of the nation's foremost aircraft designers, Issac Machlin Laddon has been named vice president and general manager of Consolidated Aircraft Corp., with whose engineering and production progress he is credited considerably. Connected with aviation since 1917, Mr. Laddon began his work in the experimental department of Cadillac Motor, following graduation from McGill university. He was called to Washington by the engineering division of the Air Service the following

year, and almost immediately was sent to McCook Field, Dayton, O., air service proving ground. In 1918 he was appointed chief of design branch No. 2, charged with bombardment airplane development, in which position he remained until 1927 when he joined Consolidated for the purpose of developing products other than training planes in which the company specialized previously. Until his present promotion he was vice president and works manager. Mr. Laddon is credited with the design of

CASE HISTORY No. 321 FROM OUR CORPRENE* FILES

PROBLEM: To provide a dust-and-oilproof seal between a hydraulic oil reservoir tank and its cover. Gasket must support loads up to 400 lbs. per sq. in. without shearing or extruding and must be in a form to fit the periphery of tanks of varying size without necessitating shaping or machining of the tank edge or lid.



THE hydraulic equipment manufacturer confronted with the above problem needed a gasket material which would (1) seal permanently against oil, dust, and moisture; (2) be tough enough to support a weight on the tank top of 400 lbs. per sq. in. without shearing; (3) be slightly compressible to take up irregularities in the edge of the tank and to minimize extending; (4) be flexible enough to bend around corners of ¾″ to 1″ radius without fracturing.

Moreover, there was a production problem involved. The manufacturer wanted the sealing material in continuous strip form so that it could be cut into lengths to accommodate tanks of various sizes.

SOLUTION

A CORPRENE material (DC-118)—supplied in slotted ribbon form—satisfied all the requirements. The slot positions the gasket and holds it in place during assembly. The cork in the compound provides the desired compressibility. The manufacturer simply cuts a length to fit each tank. There is no waste. Furthermore, there is no need for him to maintain a large inventory of various-sized gaskets.

"Corprene" is the trade-mark for more than two dozen cork-and-synthetic-rubber compounds having excellent sealing characteristics. Regardless of the type of equipment you make, you will very likely find the answer to your sealing problems by investigating the wide range of physical properties available in Corprene sheets, cut pieces, molded shapes, and extruded rings. For complete information, write Armstrong Cork Company, Industrial Division, 942 Arch Street, Lancaster, Pa.

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the first all-metal plane constructed in this country, and the first airplane brake installation which proved practical. He has also been granted numerous patents on airplane wheels, brake controls and complete airplane design.

♦

Paul H. Miller, associated with Carboloy Co. Inc. engineering department for the past eight years, succeeds James R. Longwell as chief engineer. Mr. Longwell has been made factory manager.

♦

Harmon S. Eberhard, well-known as an expert in army tank, tractor and gun-carrying devices, was recently named vice president of Caterpillar Tractor Co. He first entered the tractor field in the employ of the company's predecessor, and in 1925 was transferred to the engineering department. Three years later he became assistant general chief engineer and subsequently chief engineer in charge of research.

♦

Dr. C. F. Rassweiler, who last June joined Johns-Manville Corp., New York, as director of research, has been made vice president. He will continue in charge of research and development activities.

♦

F. P. McKegney has been promoted from assistant chief engineer to chief engineer of Air Reduction Co., New York.

♦

Samuel S. Bradley was recently re-elected chairman of the board of the Manufacturers' Aircraft association. Frank H. Russel of National Aviation Corp., will continue as president, while John A. Sanborn was re-elected general manager.

♦

F. F. Seaman, general manager of Robbins & Myers Inc., hoist and crane division, Springfield, O., has been re-elected chairman of the Electric Hoist Manufacturers' association. A. S. Watson was made vice chairman. Mr. Watson is vice president of Detroit Hoist & Machine Co.

♦

Arthur B. Schultz has become associated with All-American Aviation Inc., Wilmington, Del., as chief engineer of the experimental and development department. Until recently he was chief engineer of Kermath Mfg. Co., Detroit.

♦

L. H. Grisell was recently made chief engineer of Kermath Mfg. Co. Mr. Grisell was chief engineer with Scripps Motor Co. before his present appointment.

♦

Raymond H. Rice, previously chief engineer of North American Aviation Corp., has been named vice president in charge of engineering. He first became associated with the aircraft industry as a junior engineer in the Army Aircraft branch at Wright Field, and has been with the North American Aviation Corp. since 1935.

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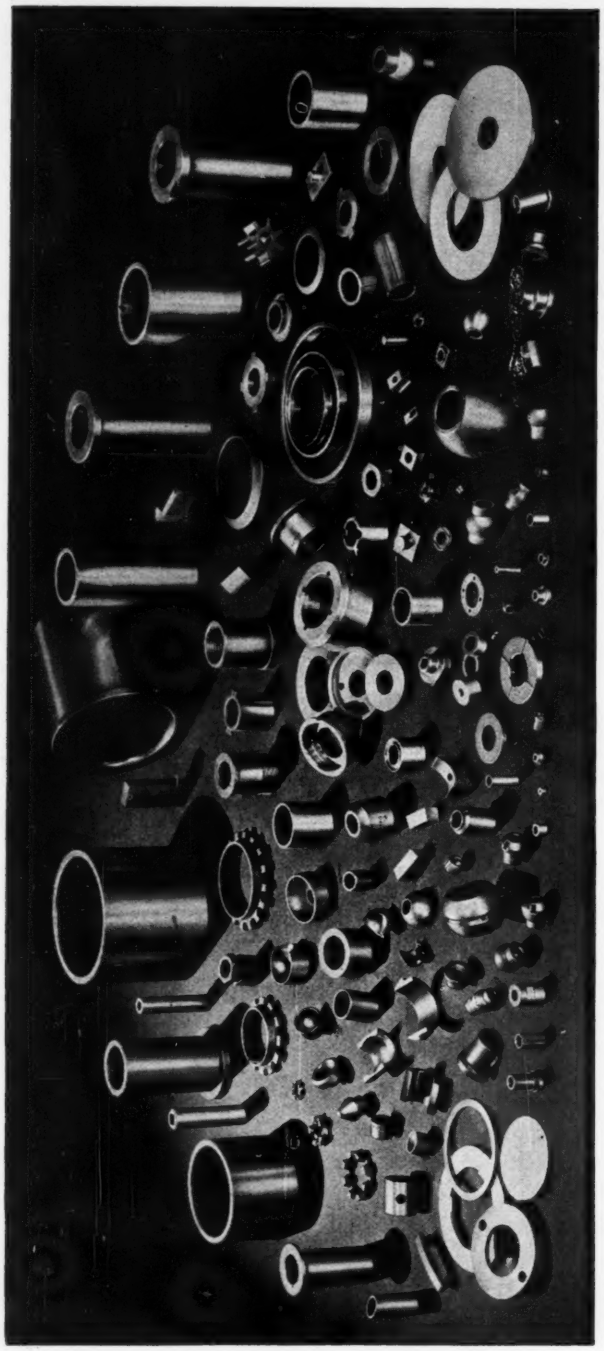

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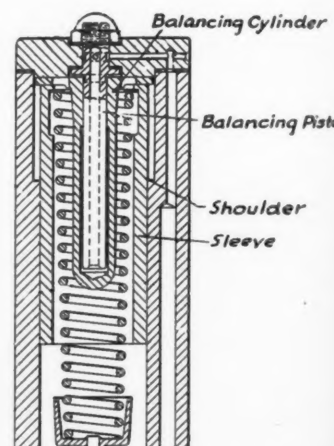
Noteworthy PATENTS

Simplifies Hydraulic Brake System

COMBINATION of a control valve and a pressure reducing or regulating valve has been achieved in one unit as disclosed in a patent assigned to the Jeffrey Manufacturing Co. The mechanism has been developed for use on the braking systems of such heavy vehicles as mine locomotives, trucks, tractors, etc.

Fluid, under a pressure considerably higher than that needed for the braking system, is introduced into the inlet port of the control valve from an auxiliary fluid accumulator. The position of the mechanism as illustrated corresponds to the fully released condition of the brakes. The high pressure fluid is therefore trapped between the two enlarged sections of the control valve, rendering the system in perfect balance. In the event that fluid leaks past the lower end of the control valve into the chamber above the plug, it immediately drains to the reservoir. The same is true of leakage occurring above the control valve.



Hydraulic accumulator safeguards pump and supplies high pressure fluid to control valve

When the brakes are to be applied, the sleeve which is a free fit in the housing, is moved down-

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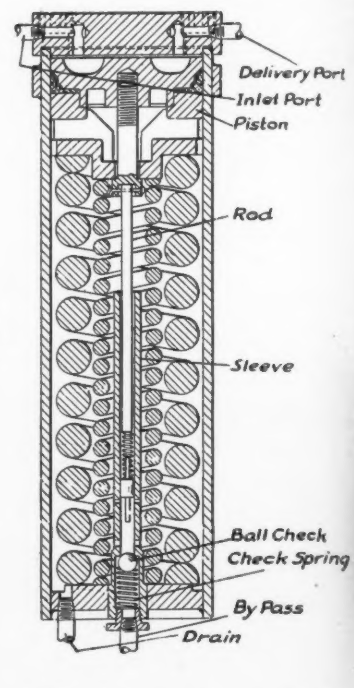
ward. This is accomplished by means of rack teeth formed in the sleeve engaging a corresponding pinion operated by a convenient crank or lever. The rack and pinion mechanism are not shown. Since the pressure in the chamber containing the conical spring is that of the reservoir, or substantially atmospheric, no other packing than an oil seal is required about the pinion shaft. As the sleeve is lowered to apply the brakes the conical spring exerts a downward thrust on the control valve piston. The control valve piston moves downward correspondingly until a passageway is opened to the delivery port which supplies pressure fluid to the braking system.

As fluid, under pressure, flows to the brake jacks, it flows also through the passageway across the bottom of the mechanism, thence through the vertical passage formed in the right side of the housing through the horizontal hole in the upper cap and down through the central drilling in the balancing piston. Constituting a substantially fluid-tight fit around the balancing piston is a balancing cylinder closed at the lower end. Since the cross-sectional area of the balancing piston is equal

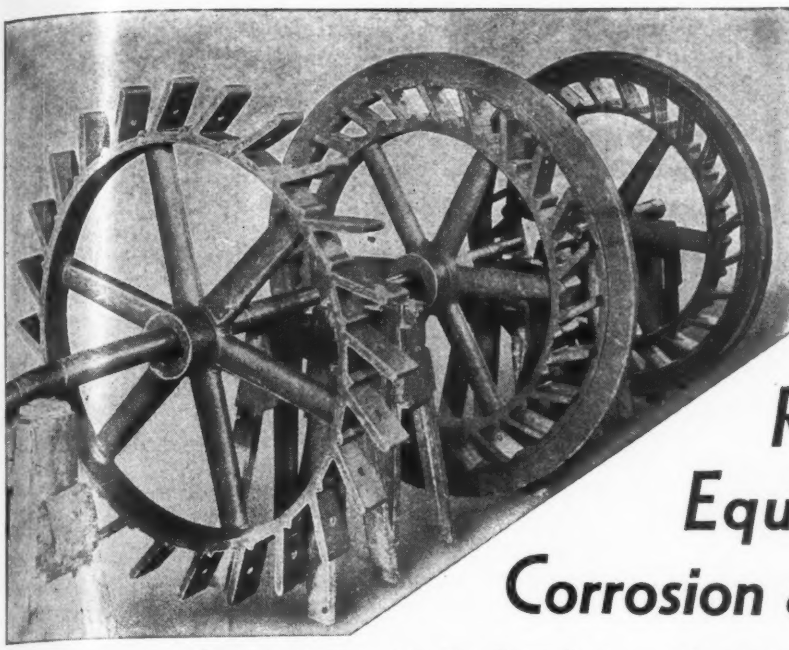


Valve acts both as pressure regulator and controller in hydraulic braking system

to that of the control valve, complete pressure balance of the latter is attained. This is accomplished by means of the downward thrust on the conical spring imparted by the flange of the balancing cylinder and applied to the spring retaining cup which is an integral part of the control valve.

If the sleeve is lowered so that its upper end is in any position above the shoulder formed in the valve housing, it is evident, because of the complete pressure balance of the system, that it will remain in that position without requiring any operator effort. A predetermined pressure corresponding to such a position will then be supplied through the delivery port to the braking system.

If however, the sleeve is lowered so that its upper end drops below the shoulder in the housing,



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the flange of the balancing cylinder will rest upon this shoulder rather than upon the sleeve. In this position the balancing effect of the mechanism is negated and the high delivery pressure exerts an unbalanced force against the bottom of the control valve piston tending to raise it cutting off the flow. Thus a manual effort, on the part of the operator, is required to hold the valve in this position. If the operator releases the handle the sleeve will immediately return to the shoulder in the housing corresponding to the maximum normal operating pressure.

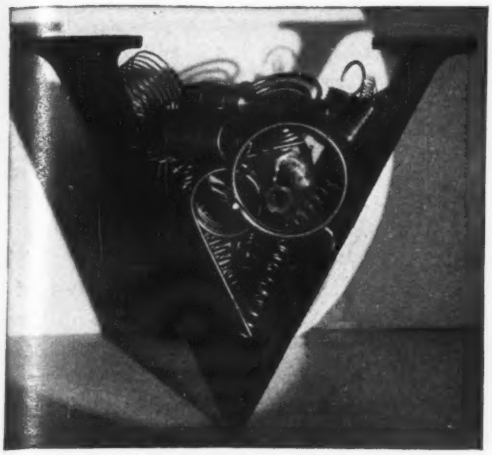
Accumulator Has Safety Features

The previously mentioned accumulator shown in the second illustration is of interest because of the automatic safety overload mechanism which it includes. It is the function of this accumulator to supply high pressure fluid to the combination control and reducing valve mentioned in the foregoing.

The discharge line of a hydraulic pump is connected to the accumulator at both the inlet port and the pipe connection at the by-pass. A drain connection leading to the sump or reservoir is also provided. In the position shown, the fluid pressure within the accumulator is equal to zero. When the pump is started, hydraulic fluid flows through the inlet port and builds up pressure over the top of the piston. Under the influence of this pressure, the piston is moved downward compressing the two concentric helical springs. A check valve installed in the inlet pipe and not shown in the illustration prevents fluid from backing up into the hydraulic pump. The spring-loaded piston serves to maintain a constant supply of hydraulic fluid under pressure to the control and reducing valve. As the pressure over the piston builds up to a predetermined value (for example, 900 pounds per square inch) the automatic by-pass mechanism built into the accumulator goes into operation. Attached to the under side of the piston, is a long rod which fits loosely in a guide sleeve integrally attached to the lower head of the accumulator. Supported by a light compression spring on the under side of this rod, is a small piston. As the accumulator piston is forced downward the axial extension on the lower side of the small piston comes into contact with the ball-check valve. Because the light spring is insufficient to thrust the ball-check valve open against the pressure of hydraulic fluid from the pump applied to the underside of the ball, the by-pass is not effective until the light spring is compressed to the point where the upper axial extension is in contact with the rod.

Combined Slip and Positive Clutch

EMBODYING the combined functions of a slip-clutch and positive, or dog-clutch, a mechanism discussed in a patent assigned to Felt & Tarrant Manufacturing Co. has been designed for use in the operating mechanism of calculating machines. The driving member of such a mechanism has a re-



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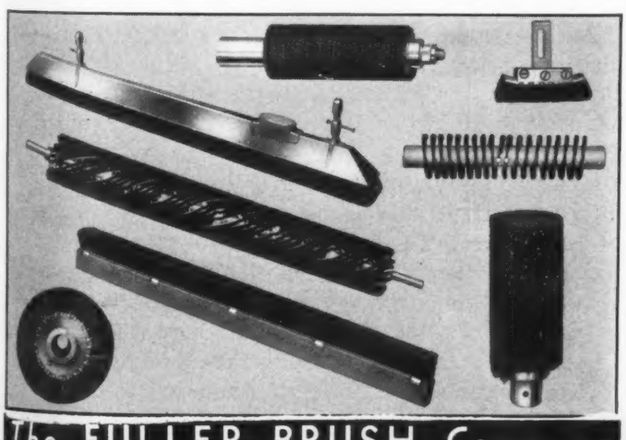

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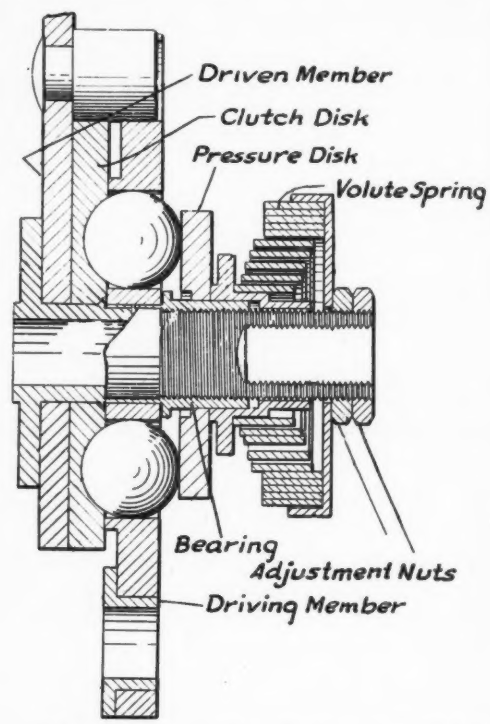
ciprocating motion about its shaft, the complete arc of motion being less than a full circle.

The entire mechanism is mounted on, and is concentric with, a single shaft. The driving member comprises a disk with an attached handle. Confined in a series of holes in the driving member disk are a number of balls. These balls engage in recesses in the clutch disk which is fastened integrally to the driven member. These recesses are elongated circumferentially and are deeper on one end than the other. As illustrated, the balls are engaged in the deepest part of the recess and the clutch is in full driving engagement.

In order to maintain the balls in this engagement, a volute spring is employed. The engagement thrust of this spring is capable of being varied by means of the two adjustment nuts on the threaded end of the shaft.



Volute spring permits adjustment of clutch engaging pressure without wide load variations in the drive

As the driving member is rotated in a clockwise direction through a given angle, it is apparent that the engagement of the ball in the clutch disk will affect a similar rotation of the driven member. However, as the driven member comes up against a stop, the driving member may continue to rotate in which case the balls will roll to the right, out of their recesses, thereby compressing the spring.

On the return stroke of the driving member, an arm on this member will engage a stud fastened to a corresponding arm on the driven member. Thus the two members will be rotated in unison in a counter clockwise direction when viewed from the right. This motion then continues until both are in the initial positions with the balls, under the influence of the spring, returned to the deepest portion of the recesses in the clutch disk.

Pump Design Affords Wide Scope in Application

(*Concluded from Page* 58)

ameter required to generate a certain head at a given speed.

Efficiency and shaft horsepower requirement of a pump are further affected by internal leakage losses through the wearing rings, disk friction of the exterior of the impeller, and mechanical losses in bearings and packing boxes. Internal losses are kept down by close clearance and labyrinth rings. The minimum disk friction for a given operating condition will be obtained if the highest practicable speed is selected, since the disk friction for constant circumferential velocity U_2 increases approximately with the square of the impeller diameter.

For a pump having a flat head-capacity characteristic the horsepower required at no delivery is close to half the power required at full delivery. It increases gradually to the rated delivery and flattens out to a maximum value for further increase in capacity. For a steeper head-delivery characteristic, the power taken at no delivery is relatively higher, corresponding to the higher head developed at this point, and for extremely steep head-delivery curves, such as those for mixed-flow pumps, the power at shut-off is equal to or greater than the power at the rated capacity.

Possible to Predict Operation

When the characteristic curve of a pump at a certain speed n is given, it is possible to predict the operation at a different speed n_1 fairly accurately. Thus, if the speed is changed from n to n_1 the tip velocities U_1 and U_2 change in the same proportion, n/n_1, and for a shockless inlet the velocity C_1, *Fig.* 2, must likewise change in the same proportion. As C_1 is a measure of the capacity Q_1, for the rated point $Q_1 = Qn_1/n$, or the capacity changes in direct proportion to the speed. All velocities in the outlet diagram change in the same proportion. As in Equation 2, $H = U_2Cu_2/g$ and as both U_2 and Cu_2 change in direct proportion to the speed, the corrected head is $H_1 = (n_1/n)^2$, or the head for the rated point changes as the square of the speed. The same correction may be applied for any point on the characteristic curve, as well as at the rated point.

At each corresponding point the pump will have approximately the same efficiency at the corrected speed as at the original speed, within reasonable limits. As the capacity changes directly with the speed, and the head with the square of the speed, and as the efficiency remains the same, the power for each corresponding point changes approximately as the cube of the speed.

Surging in Springs

(*Continued from Page* 51)

neglects the transient condition which is of no concern here. The assumption represented by Equation 27 is justified, since for a forced vibration of circular frequency ω_n all parts of the spring must vibrate at this frequency.

Boundary conditions are: For $x=0$, *Fig.* 5, $y=0$ regardless of time t, since one end of the spring is assumed fixed in space. For $x=l$, $y=c_n \sin \omega_n t$ since the other end of the spring is assumed to have a harmonic motion of amplitude c_n produced by the harmonic of Equation 2 which is in resonance with the natural frequency.

A solution of Equation 9 which satisfies these boundary conditions has been obtained by Hussmann[4]. For small values of damping such as occur in practical springs the solution reduces to the relatively simple form

$$y_{max}=\frac{2c_n f_n}{b\lambda} \sin\,(2\pi f_n t+\phi_n) \quad (28)$$

where $y_{max}=$ maximum amplitude of motion at any coil in the spring, ϕ_n is a given phase angle and λ is the order of vibration. From this solution, for small damping, the maximum variable stress is

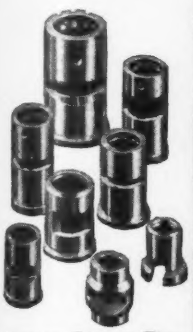
$$\tau_v=\tau_{st}\frac{2\pi f_n}{b} \quad (29)$$

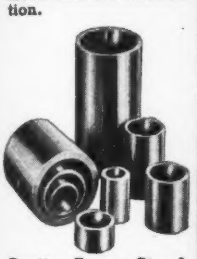
In this the stress τ_{st} is the static stress induced by compressing the spring by an amount c_n. This equation indicates that the variable stress τ_v, *Fig.* 7, is inversely proportional to the damping factor b and directly proportional to the frequency f_n and stress τ_{st}. The latter, in turn, is proportional to the amplitude c_n of the particular harmonic in resonance.

Tests have shown that the damping factor b in actual springs may be low enough that a magnification of 100 to 300 times occurs, i.e., τ_v may be 100 to 300 times τ_{st}. It has also been found[4] that the damping factor b varies with the amount of initial compression of the spring and that it increases with the amplitude c_n of the harmonic in resonance. This is reasonable because, at low amplitudes, the internal damping of the spring material will be lower due to hysteresis. Also for extremely high initial compressions, higher values of b are found due to damping caused by impact between turns. At medium initial compressions, values of the damping factor are lower while at very low compressions these values again rise because of damping from the clashing and lifting of the end turns from the supports.

Values of b varying from about 1 sec^{-1} at lower amplitudes of vibration to 10 sec^{-1} at the higher amplitudes have been obtained in tests[4], most values being between 2 and 4.

As an example of the use of Equation 29 it may





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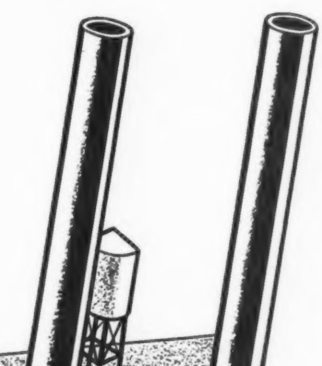

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be assumed, for example, that the lowest natural frequency f_n of the spring is equal to 200 cycles per second and that the camshaft speed is 1200 revolutions per minute or 20 revolutions per second. This means that resonance between the natural frequency of the spring and the 10th harmonic of the valve lift curve may occur. Assuming that tests on springs under similar conditions have shown a damping factor of $b=5$ sec^{-1}, Equation 29 becomes

$$\tau_v=\frac{2\pi f_n}{b}\tau_{st}=\frac{2\pi\times 200\tau_{st}}{5}=251\tau_{st}$$

Letting τ_0 be the stress due to compression of the spring by an amount equal to the valve lift then, if the alternating stress due to the 10th harmonic of the valve lift curve is .002 τ_0 (as found from harmonic analysis), the alternating stress due to resonance with this harmonic will be $251\times .002\tau_{st}$ or about $.5\tau_0$. This means that in this case the stress range will be increased to 2 times its value with no vibration.

From *Fig.* 7 the total stress range in the spring is

$$\tau_r=\tau_0+2\tau_v \qquad (30)$$

If τ_1 is the stress due to initial compression of the spring with the valve in the closed position, the range in stress will be from a minimum value $\tau_{min}=\tau_1-\tau_v$ to a maximum value $\tau_{max}=\tau_1+\tau_0+\tau_v$. By comparison with endurance diagrams of the type given in a previous article[10] the relative margin of safety of the spring against fatigue failure may be estimated.

A typical resonance curve similar to those obtained by actual tests on valve springs is shown in *Fig.* 8. In this curve the amplitude of oscillation of the middle coil of a valve spring is plotted against camshaft speed. It is seen that this curve consists of various peaks spaced at intervals, each peak being due to a definite harmonic in the valve lift curve and indicated by the number shown. Thus the peak marked 10 is due to the 10th harmonic of the valve lift curve, i.e., to a vibration frequency of $10\times 20=200$ cycles per second for a camshaft speed of 1200 revolutions per minute. The amplitudes of these peaks vary since the amplitudes, i.e., values of the c's of Equation 2, of the various harmonics are different.

In the design of springs subject to rapid reciprocating motion the following expedients are often helpful: (1) Use of a high natural frequency, (2) change in pitch of coils near end of spring, (3) where possible, avoidance of resonance between a natural frequency of the spring and an exciting frequency, (4) change in shape of cam so as to reduce the magnitude of the harmonics of the valve lift (or spring-deflection) curve within certain speed ranges of practical importance. By using methods given, in conjunction with test results, estimates of stress ranges in actual springs under vibration conditions can be made and in this way margin of safety against failure determined.

[10] "Spring Materials," MACHINE DESIGN, Page 46, October, 1940. See also "Permissible Stress in Small-Sized Helical Springs"—F. P. Zimmerli, *Engineering Research Bulletin* No. 26, University of Michigan.

Business and Sales Briefs

HAVING had experience in various headquarter sales sections, Robinson S. Kersh, an industrial sales engineer, has been appointed manager of the Houston, Texas, sales office of Westinghouse Electric & Mfg. Co., East Pittsburgh. Previous to his present appoinment Mr. Kersh served in the machinery electrification and aviation section at East Pittsburgh.

From the Milwaukee office of Ampco Metals Inc., J. E. Heuser has been transferred to the Cincinnati office to assist J. E. Cook.

Associated with American Chain division of American Chain & Cable Co. Inc., York, Pa., for several years, C. A. Puckett has been appointed sales manager of the weldless chain department.

Announcement has been made of new addresses for district offices of Lincoln Electric Co., Cleveland. These are: 19 North Ellison street, Oklahoma City, Okla., with R. L. Looney as manager; 733 North Van Buren street, Milwaukee, with F. C. Archer as district manager; 1111 James building, Chattanooga, Tenn., with R. M. Daniels in charge; 323-325 East Twenty-third street, Chicago, with G. E. Tenney continuing as manager; 521 South Fifteenth street, Omaha, Nebr., with Fuchs Machinery & Supply Co., as its representative; and 246 Wiltshire boulevard, Dayton, O., with R. P. Sharer as manager in charge.

Large scale expansion of the plant of American Agile Co., Cleveland, has been announced. Growing popularity of the company's products plus a nationwide response to the application of welding in the country's armament program are the basic reasons for the expansion. The company will continue to add new items to its line of welding electrodes.

Another 15,000 square feet of floor space and additional modern screw equipment have been acquired by Manufacturers Screw Products, 212-222 West Hubbard street, Chicago, manufacturer of screws, bolts, washers and rivets for ships, tanks, planes, gun mounts and many other essential defense equipment.

John W. Clarke Co., 327 South LaSalle street, Chicago, has been appointed sales and engineering representative of Struthers Dunn Inc. for its line of relays and timing devices in the state of Wisconsin, the upper peninsula of Michigan, Northern Illinois, Chicago and neighboring counties of Indiana, and the eastern border counties of Iowa.

Acquisition of the Cunningham Controls Co. of Burbank, Calif., has been announced by the General

Controls Co., Glendale, Calif. The Cunningham company has been actively engaged for several years in manufacturing temperature controls for transport planes, and also has specialized in all forms of temperature control systems, with particular emphasis toward airplane cabin heat controls. The General Controls Co. has been supplying aircraft equipment, particularly lightweight magnetic valves, and the addition of the temperature control completes its line of aircraft equipment. L. L. Cunningham, for the previous seventeen years a temperature control engineer with various leading manufacturers, joined the General Controls organization in an engineering capacity and will concentrate on temperature control engineering and design.

♦

Aircraft control devices are now being built by General Electric Co. These include solenoids, relays, contactors, and pressure and limit switches, some of which are also applicable to tank and other installations.

♦

Demand for Copperweld Steel Co. stainless, Nitralloy and other special steels has made it necessary for the company to create a new department devoted to the sales of tool, stainless and special steels. Paul Keller has been placed in charge of these sales. He was formerly manager of the company's Cleveland sales office.

♦

Previously sales manager for the aeronautical industry of Reynolds Metals Co., T. A. Lynch has been elected a vice president. The company is a fabricator of aluminum parts as well as manufacturer of virgin aluminum.

♦

Walter H. Blocher has been made assistant manager of sales, Republic Steel Corp.'s alloy steel division, Massillon, O. Following his graduation he joined Central Alloy Steel Corp., and was in the Indianapolis office of this company when it became a part of Republic.

♦

Formerly mechanical engineer at the railway steel spring division of American Locomotive Co., Bennett Burgoon Jr. has been appointed new district representative in Western Illinois and Iowa by McKenna Metals Co., Latrobe, Pa. His offices will be at 917 Talcott building, Rockford, Ill.

♦

Well known as manufacturer of vertical pumps, Pomona Pump Co., Pomona, Calif., has purchased the Westco pump division of Micro-Westco Inc., Bettendorf, Ia. The newly-acquired business will be operated as the Westco division of Pomona Pump Co., 2621 Locust street, St. Louis.

♦

Offices of Sterling Plastics Co., custom molders, have been moved into the company's new plant at 1140 Commerce avenue, at Vaux Hall road, Union, N. J.

♦

Originally formed to uncover new uses for seamless steel tubing, the Seamless Steel Tube institute, Gulf building, Pittsburgh, is now taking specific steps to disseminate technical data and useful practical information on tubing, together with help members of the organization may be able to give

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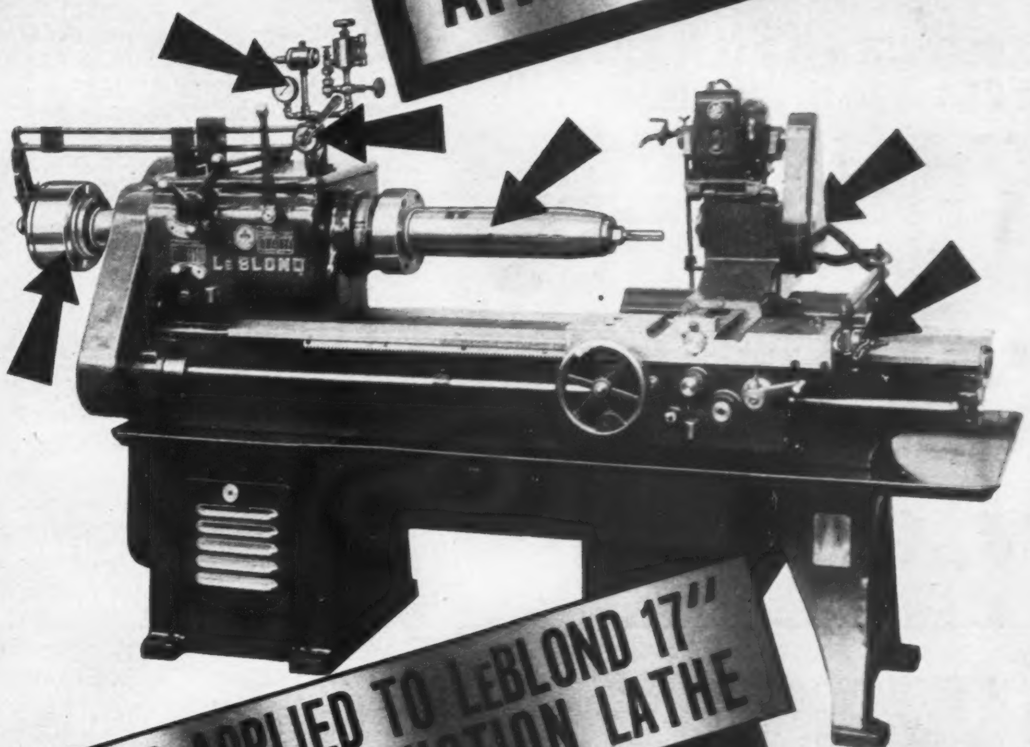
AIR EQUIPMENT



AS APPLIED TO LeBLOND 17" RAPID PRODUCTION LATHE

THIS LeBlond Rapid Production Lathe is designed for the centering operation of shells. It is equipped with a "LOGAN" Model "R" Double Acting Rotating Type Air Cylinder to supply power efficiently, economically to a "LOGAN" 6-Jaw Expanding Mandrel. A Model "F" Non-Rotating Type "LOGAN" Air Cylinder (not shown in illustration) pulls the cross-slide quickly out of the way to load and unload the machine. In addition, this LeBlond Lathe is also equipped with a "LOGAN" Reducing Valve, Lubricator and Pressure Gauge Unit and "LOGAN" Model "TL" Air Control Valves. This "LOGAN" Air Equipment provides rapid, positive action and rigid support of the work. "LOGAN" Representatives and "LOGAN" Engineers will be glad to make recommendations on your problems.

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users of tubing to the end that designers, engineers and fabricators who specify or use tubing may have the maximum help possible in saving time and producing more war materiel. In other words, the accent is now upon service to tube users rather than a search for new users.

♦

Gray Iron Founders' society has arranged to furnish expert technical advice on uses to which gray iron castings can be applied in maintaining rigid conservation of materials necessary to the war economy. A. J. Edgar, until recently in charge of the gray iron, steel and nonferrous foundries of General Railway Signal Co., has come to the society with a well-rounded knowledge of gray iron castings and their applications in industry. Mr. Edgar will serve on the staff of W. W. Rose, executive vice president of the society, at the society's Washington office, 1719 K street, Northwest.

They Say . . . .

"We have failed to realize that the decency of democratic instincts is no weapon against the practical barbarism of the dictators. . . . The day of bluffing, or of 'talking' a good war is past. We are down to the naked truth that planes, ships and tanks are about the only things that count. . . . If we will take this last sentence literally and make it our national objective we will win the war. If we continue to take liberties with it, we will go down in defeat."—E. L. Shaner.

♦

"The war has made necessary the full industrial mobilization of America. It means drastic changes in our existing industrial economy. We must think only in terms of outproducing a powerful enemy; and anyone who underestimates the ability of the enemy to produce is kidding himself."—Donald M. Nelson.

♦

"Some of the already heavy requirements placed upon us have just recently been more than doubled, and seven day a week operation on a twenty-four hour basis will be the rule whenever this will result in greater output of the many weapons we are making for those fighting our battles in the front lines."—Charles E. Wilson.

♦

"Complexity of a part is just a confession of poor design."—Dr. A. V. deForest.

Meetings and Expositions

April 15—

Packaging Machinery Manufacturers institute. Semiannual meeting to be held at Hotel Astor, New York. H. L. Stratton, 342 Madison avenue, New York, is secretary.

April 16-17—

National Petroleum association. Semiannual meeting to

Check

THE ITEMIZED INDEX!

All drive and control developments in this issue are listed on page 5.

Every reference to materials as used in machinery can be located through page 5.

Individual design problems are classified for maximum utility, on page 5.

•

In every issue of MACHINE DESIGN all problems involving the application of parts, materials, mechanisms—in short, every detail covered by that issue is carefully *itemized.*

Constant reference is made to the Itemized Index by alert engineers as a routine part of their design procedure.

SEE PAGE 5!

be held at Cleveland hotel, Cleveland. M. C. Mallon, 930 Munsey building, Washington, is secretary.

April 20-24—

American Foundrymen's association. Forty-sixth annual convention and exhibition to be held at Public Auditorium and Exhibition hall, Cleveland. R. E. Kennedy, 222 Adams street, Chicago, is secretary.

April 20-24—

American Chemical Society. Spring meeting to be held at Hotel Peabody, Memphis. C. L. Parsons, 1155 Sixteenth street, Washington, is secretary.

April 22-24—

Petroleum Industry Electrical association. Annual meeting and exhibition to be held at the Washington-Youree hotel, Shreveport. J. F. Collerain, P. O. Box 2412, Houston, is secretary.

April 27-28—

Association of Iron and Steel Engineers. Spring conference to be held at the Royal Connaught hotel, Hamilton, Ont., Canada. J. L. Miller, Empire building, Pittsburgh, is secretary.

April 27-May 1—

American Mining Congress. Nineteenth annual coal convention and exposition to be held in Cincinnati. J. D. Conover, Munsey building, Washington, is secretary.

May 4-6—

American Supply and Machinery Manufacturers' association. Convention to be held at Hotel Traymore, Atlantic City. R. Kennedy Hanson, 1108 Clark building, Pittsburgh, is general manager.

May 10—

National Electrical Manufacturers association. Annual meeting to be held at The Homestead, Hot Springs. R. J. Blais, 155 East 44th street, New York, is convention manager.

May 11-13—

American Institute of Chemical Engineers. Thirty-fourth semiannual meeting to be held at the Statler hotel, Boston. S. L. Tyler, 50 East 41st street, New York, is secretary.

May 11-13—

American Gear Manufacturers association. Twenty-sixth annual convention to be held at Hotel Hershey, Hershey. J. C. McQuiston, 602 Shields building, Wilkinsburg, Pa., is secretary.

June 8-11—

American Society of Mechanical Engineers. Semiannual meeting to be held in Cleveland. C. E. Davies, 29 West 39th street, New York, is secretary.

June 17-19—

American Society of Mechanical Engineers. Oil and Gas Power division meeting to be held in Peoria. C. E. Davies 29 West 39th street, New York, is secretary.

June 22-25—

American Society of Agricultural Engineers. Annual meeting to be held at Schroeder hotel, Milwaukee, Raymond Olney, Box 229, St. Joseph, Mich., is secretary.

June 22-26—

American Institute of Electrical Engineers. Summer convention to be held at the Drake hotel, Chicago. H. H. Henline, 33 West 39th street, New York, is national secretary.

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★ ★ ★ ★

NEW MACHINES—
And the Companies Behind Them

(For illustrations of other outstanding machinery, see Pages 120-121)

Air Conditioning

Air heating unit, Air Conditioning Div., Gar Wood Industries Inc., Detroit.

Domestic

Glass-encased alarm clock, William L. Gilbert Clock Corp., Winstead, Conn.
Refrigerators, Westinghouse Electric & Mfg. Co., Mansfield, O.
Enamel pressure cooker, Vischer Products Co., Chicago.
Combination radio-television set, General Electric Co., Bridgeport, Conn.

Finishing

Machine for spray-painting incendiary bombs, DeVilbiss Co., Toledo, O.
Portable air-driven sander, Sterling Tool Products Co., Chicago.
Automatic spray machine for coating and drying small shells, Eclipse Air Brush Co., Newark, N. J.
Metal sheet coating machine, Chas. E. Francis Co., Rushville, Ind.
Portable electric sander, Detroit Surfacing Machine Co., Detroit.
Dipping-whirling machine, Ronci Machine Co., Providence, R. I.
Electric eye shell sprayer, Eclipse Air Brush Co. Inc., Newark, N. J.
Oil and water extractor, Binks Mfg. Co., Chicago.

Food

Air-powered slow speed mixer, Eastern Engineering Co., New Haven, Conn.
Improved hand-pack filler, Food Machinery Corp., Hoopeston, Ill.
Milk bottle filler and capper, Pfaudler Co., Rochester, N. Y.
Drying system, Industrial Associates Inc., New York.
Power flour mills, Arcade Mfg. Co., Freeport, Ill.
Chlorine producing machines, The Valhalla Co., Chicago.

Forging

Rotary forge furnace, Rhode Island Tool Co., Providence, R. I.

Industrial

Hydro-foam dust collector, R. C. Mahon Co., Detroit.
Optical comparator, Portman Machine Tool Co., Mt. Vernon, N Y.
Automotive double ribbon type mixer, H. K. Porter Co., Pittsburgh.

Materials Handling

Lightweight fork truck, Clark Tructractor Div., Clark Equipment Co., Battle Creek, Mich.
Electric tractor, Mercury Mfg. Co., Chicago.
Lift truck, Crescent Truck Co., Lebanon, Pa.

Metalworking

Six-inch horizontal boring machine, Simmons Machine Tool Corp., New York.
*Vertical milling machine, Reed-Prentice Corp., Worcester, Mass.
High-speed milling machine, Pope Machinery Corp., Haverhill, Mass.
*Thread and form milling machine, Gordon-R Co., Royal Oak, Mich.
Horizontal carbon and diamond borer, Glern & Anholtt Tool Co., Detroit.
*Production thread miller, The Lees-Bradner Co., Cleveland.
*Semiautomatic miller, Snyder Tool & Engineering Co., Detroit.
*Surface grinder, The Thompson Grinder Co., Springfield, Ohio.
*Hypoid grinder, Gleason Works, Rochester, N. Y.
*Hydraulic bending press, E. W. Bliss Co., Brooklyn, N. Y.
Tap reconditioner, Detroit Tap & Tool Co., Detroit.

(Concluded on Page 234)

Office

Ten-key electric adding-multiplying machine, Barrett Adding Machine Div., Lanston Monotype Machine Co., Philadelphia.

Optical

Optical grinding machine, George Scherr Co. Inc., New York.

Quarry

Rock drill, Ingersoll-Rand Co., Phillipsburg, N. J.

Restaurant

Deep fat fryer, thermostat controlled, Sheridan Specialty Co., Chicago.

Textile

Finishing range, Reliance Dyeing & Finishing Co., Covington, Ky.

Double tricot machine, Alfred Hofmann Inc., West New York, N. J.

Concentrated air cloth dryer, Bridges-Wilson Corp., Arlington, Mass.

Welding

Two-wheeled, lightweight welder-trailer, Hobart Bros. Co., Troy, O.

Heavy-duty transformer for automatic welding, Glenn Roberts Co., Oakland, Calif.

Horizontal welder for shells, Federal Machine & Welder Co., Warren, O.

*Automatic shell welder, Thompson Gibb Electric Welding Co., Lynn, Mass.

Woodworking

Strippers, Strippit Corp., Buffalo, N. Y.

Power squaring shears, Niagara Machine & Tool Works, Buffalo, N. Y.

*Illustrated in pictorial spread, Pages 120-121.